PREFACE—FIFTH EDITION

The remarkable progress in the oil engine industry in recent years has necessitated great enlargement of, and considerable additions to, this treatise in order that it may include all branches of an important and interesting subject.

Arrangement in two parts has for this reason been decided upon. In Part I the modern high pressure solid injection, the air blast design and the modern two cycle engines are discussed, while in Part II the earlier types, their history and development are treated of.

This work, therefore, will be found thoroughly comprehensive, as it traces the activities in this art from its inception to the present time. It chronicles each development that has occurred and also sets forth the many improvements made in recent years. The descriptions of the older engines have been retained as this information forms a reference of great value, it is indispensable wherever such engines are still in service and is also useful for purposes of comparison.

The profuse supply of detailed illustrations will serve to simplify and make plain the descriptive parts of the text.

The appendix in the fourth edition relating to the Diesel Engine is withdrawn as that matter now forms the subject of a separate treatise devoted to Diesel Engines alone.

The author hereby records acknowledgment to the various manufacturing firms in Europe and in the U. S. A., who have kindly placed drawings, etc., at his disposal and to them and others who have assisted in the preparation of this work appreciation is tendered. In each instance where extracts have been made for illustrations or text the origin is referred to and gratitude is hereby expressed for permission to follow this procedure.

CONTENTS. PART I

CHAPTER I

INTRODUCTORY—FIRST PRINCIPLES

Comparison of Efficiencies of Early and Modern Hot Surface Types—Reasons for Higher Economy—Advantages—Starting—Compression Pressures—Capacities—Ignition—Thermodynamics of the Oil Engine—Definitions—Air Standard Diagram—Compression Cycles—Thermal Efficiencies. Heat Losses—Cycles—Comparison of Two and Four Cycle—Volumetric Efficiency.. 1

CHAPTER II

DESIGN AND CONSTRUCTION OF PARTS—VAPORIZERS—SPRAYERS

Range of Pressures—Formulae for High Pressure Engines—Main Frame and Cylinder—Cylinder Liner—Cylinder Casing—Cylinder Head Bolts—Combustion Space. Sprayers—Function—Nozzles. Dimensions. Pulverizers—Various Designs of Sprayers for 4 Cycle and 2 Cycle Engines—Regulation of Speed—Different Methods—Oil Pump—Connecting Rod—Connecting Rod Bolts—Air Compressors—Capacity—Valves—Air Inlet and Exhaust Valves. Exhaust Ports. Pistons. Dimensions of Same—Crankshafts—Dimensions and Comparison of Various Types—Built Up Crankshafts—Crankshaft Metal Specifications—Lloyd's Rules for Marine Crankshafts—Auxiliaries—Balancing Reciprocating parts—Stroke Ratio—Piston Speed—Flywheels—Parallel Operation 15

CHAPTER III

TESTING—INSTALLATIONS—OPERATION AND CORRECTION OF OIL ENGINES

Code of Tests of Am. Soc. Mech. Engrs.—Object—Operating Conditions—Duration—Starting and Stopping—Records—Calorific Tests and Analyses—Heat Consumptions—

H. P. and Efficiency—Heat Balance, Data and Results —Valve Settings—Four Cycle—Two Cycle—Installation—Arrangement of Engine Room—Heat—Light—Ventilation—Floor—The Foundation of Engine—Concrete Mixture—Weight of Foundation—Size—Insulation—Exhaust Pipes—Dimensions—Arrangement of—Test Cock—Air Inlet Pipes Sizes—Cooling Water Arrangements—B. t. u. Carried Away—Various Methods of Cooling—Enclosed Cooling System—Temperatures —Quantities of Cooling Medium Required—Instructions—Prevention of Freezing—Silencer Dimensions—Operation and Connection—General Remarks—Instructions—Principles of Operation—Requirements for Proper Ignition—Oil Supply to Cylinder—Too Rich and Too Poor Mixtures—Indications—Air Supply—Four Cycle and Two Cycle Types—Oil Supply to Vaporizer—Combustion Space—Knocking and How to Remedy—Deficient Power—Causes of—Piston Blowing—Carbonized Piston Rings—Overheating—Uneven Expansion—Reboring Cylinder—Leakage of Air Inlet and Exhaust Valves—Investigation and Correction—Lubricants—Specifications of 42

CHAPTER IV

TWO-CYCLE OIL ENGINE—DESCRIPTION OF VARIOUS DESIGNS

Representative Types of American and European Manufacture—Beardmore Engine—Pressures of Compression —Maximum Pressures—Flexibility of Two Cycle Type—Conditions—Fuel Consumption—Fuel Injection Period—Temperature of Hot Surface. M. E. P. of Crank Case Compression Type—Separate Scavenging Air Pump—Piston Speeds—Exhaust Velocity—The Petter Engine—Fuel Injection Variation—Compression Pressure—Indicator Diagrams Tests—Marine Engine sequence of Reversal Movements—Fairbanks-Morse Engine—Three Cylinder Marine Type—Ignition—Frame-Hot Bulb—Starting—Fuel Pump—Crankshaft—Lubrication — Spray Nozzle — Bessemer Engine — Capacities—Cooling Arrangements—Vaporizer—Mercury Filling—Fuel Economy—Dimensions—Weight of Engine—The Buckeye Two Cycle Engine, Arrangement of—Sprayer—Fuel Economy—Weight—Lubrication—Worthington Vertical Type—Capacities—Crosshead and Guide—Dimensions—Fuel Economy—Test—Indicator Diagrams 63

CHAPTER V

FOUR-CYCLE OIL ENGINES—VARIOUS DESIGNS—PRELIMINARY REMARKS

Comparison with Two-Cycle Type—The Crossley Engine—Dimensions—Physical Actions in Combustion Chamber. Scavenging—Spraying—Oil Pump—Governing—Use of Heavy Crude Fuels—Tests—Fuel Economy Curves—De la Vergne "D. H." Engine—Frame—Cylinder Head—Connecting Rod—Piston—Piston Rings—Method of Governing—Pump—Sprayer—De la Vergne "S. I." Type—Method of Fuel Supply—Spray Valve—Method of Governing—Automatic Starting—De la Vergne "F. H." Type—Tests—Diagrams of Valve Movements of Each Engine—Indicator Cards—Blackstone Engine—Method of Operation—Spray Valve Mechanism—Air Compressor—Ignition—Dual Combustion—Dimensions—Pressures—Test—Indictor Cards—Hornsby Type "R" Engine—Air Inlet and Exhaust Valves and Cams—Tests—Indicator Cards—Sprayer—Governor Trials—Campbell Engine—Sizes—Pressures—Fuel Injection—Governing—Tests—Hvid. Design—Method of Working—Sprayer—Ignition—Various Diagrams—Enthropy Diagram—Heat Balance and Efficiency Curves 90

LIST OF ILLUSTRATIONS. PART I

Fig.		Page
1.	Historical diagrams	2
2.	Periods of fuel injection	3
3.	Theoretical diagram, constant volume	12
4.	De La Vergne D H sprayer	18
5.	De La Vergne S I sprayer	19
6.	Hornsby's spray valve, etc	19
7.	Ruston spray valve and pilot injection	20
8.	Mietz sprayer	20
9.	De La Vergne S vertical sprayer	20
10.	Fairbanks Morse sprayer	22
11.	Governing arrangement bypass	22
12.	Governing arrangement bypass	23
13.	Governing arrangement wedge system	25
14.	Packingless fuel supply pump	26
15.	Connecting rod	27
16.	Connecting rod bolt	27
17.	Air compressor (two-stage)	28
18.	Air compressor valves	30
19.	Exhaust valves	31
20.	Pistons	33
21.	Crankshaft (dimension)	34
22.	Four-cycle valve diagram, representative	49
23.	Two-cycle valve diagram, representative	49
24.	Four-cycle reversing diagrams	50
25.	Complete installation (four-cycle)	51
26.	Cooling water system, enclosed	55
27.	Diagrammatic views of two-cycle vaporizers..64, 65,	66
28.	Beardmore two-cycle, sectional view	67
29.	Indicator diagram, two-cycle	68
30.	Fuel comsumption curves	69
31.	Fuel injection, varying period	71
32.	Petter two-cycle, sectional view	73
33.	Petter single cylinder	75
34.	Petter single cylinder arrangement	75
35.	Petter single cylinder, sectional view	76
36.	Petter fuel comsumption curves	76
37.	Petter indicator diagrams	77
38.	Petter indicator diagrams, light spring	78
39.	Fairbanks Morse, sectional view	81
40.	Bessemer, sectional view	83

FIG.		PAGE
41.	Buckeye, sectional view	83
42.	Worthington two-cycle	87
43.	Worthington indicator card	89
44.	Worthington indicator card	89
45.	Crossley engine, end view section	91
46.	Crossley engine, side view section	93
47.	Crossley governor	94
48.	Crossley heater	95
49.	Crossley vaporizer	97
50.	Crossley vaporizer	97
51.	Crossley vaporizer	97
52.	Crossley curves of economy	99
53.	De La Vergne D H, sectional view	103
54.	De La Vergne S I, sectional view	105
55.	De La Vergne F H, valve movement diagram	106
56.	De La Vergne D H, valve movement diagram	107
57.	De La Vergne S I, valve movement diagram	108
58.	De La Vergne indicator diagrams	109
59.	De La Vergne vertical S I type	110
60.	De La Vergne S I, sectional view	113
61.	Blackstone vaporizer	115
62.	Blackstone indicator cards	117
63.	Hornsby type R, sectional view	119
64.	Hornsby type valve movement	120
65.	Hornsby type indicator cards	121
66.	Ruston modern engine, section	122
67.	Ruston fuel consumption curves	125
68.	Ruston and Diesel comparison	125
69.	Campbell twin-cylinder	126
70.	Campbell sectional view	128
71.	Campbell indicator cards	129
72.	Hvid type, indicator cards	130
73.	Hvid sectional view	133
74.	Hvid fuel consumption curves	134
75.	Hvid entropy diagram	135
76.	Hvid curves	137
77.	Hvid B.H.P. curves	137

LIST OF TABLES. PART I

Table		Page
I.	Historical Data of Various Engines. 4-Cycle Type	4
II.	Dimensions of Representative Low, High Compression and Diesel Crankshafts	36
III.	Crankshaft Proportions	37
IV.	Steel for Internal Combustion Crankshafts	38
V.	Data and Results of Engine Tests	46
VI.	Test on Petter Oil Engine	79
VII.	Crossley Oil Engine Tests	101
VIII.	Tests on Various De la Vergne Oil Engines	112
IX.	Test on 320 B. H. P. Hornsby Type R Engine	123
X.	Test on 85 B. H. P. Hornsby Type R Oil Engine	124
XI.	Test on Campbell Oil Engine with Mexican Fuel Oil	131

CHAPTER I

INTRODUCTORY—FIRST PRINCIPLES

PRELIMINARY.—The earlier designs of Hot Surface Type Oil Engines constructed in the latter part of the nineteenth century and described in Part II, while simple in construction and successful and reliable in operation with unskilled attention, yet were admittedly inefficient and had an extremely low thermal efficiency as compared with the modern Hot Surface Type Oil Engine constructed both in Europe and in United States at this writing (1922). Fuel consumption of .9 lbs. per B.H.P. hour was seldom exceeded in the former types, while in the latter a fuel consumption of .4 to .45 lbs. per B.H.P. hour is recorded with many designs. The writer has previously outlined the advance and improvement in the oil engine industry,* and Fig. 1 shows a series of indicator cards with results obtained illustrating this improvement. Prior to 1906 and while the Diesel patents were in force, the period of fuel injection in Hot Surface Oil Engines† was found almost without exception to occur during the air inlet stroke or during the first part of the compression

*See A. S. M. E. Transactions 1916.

†The term "semi-Diesel" is sometimes applied to these types, but is considered misleading and the term "Hot Surface" is used here entirely.

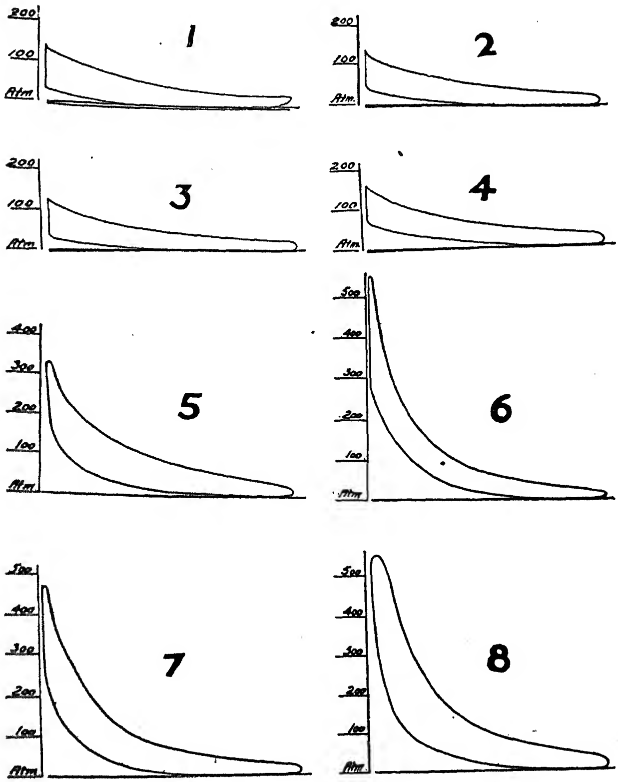


FIG. I.—VARIOUS PRESSURES, &c. OF DIAGRAMS

Diagram No.	1	2	3	4	5	6	7	8
Date	1888	1890	1905	1906	1913	1914	1915	1920
Compress. press. lbs.	20	40	46	60	169	280	260	550
Max. press. lbs.	125	120	120	168	325	550	475	550
Mean effect: press. lbs.	44	35	37.5	48	75	70	82	95
Thermal effic. %	12.8	13.5	13.8	18.	25.	29	30.5	32.5
Exhaust press., lbs.	22	20	22	30	25	25	30	40
Cyl. diam., inches	10.75	11	8.02	14.5	14	..	17	18.8
Stroke, inches	14	15	14	17	24	..	27½	28.3
Fuel consumption lbs. per B.H.P. hour	1.05	1.00	0.99	0.74	0.54	0.46	0.45	0.407

stroke. Thus during the greater part or whole of the compression stroke, the combustion space or cylinder was filled with an explosive mixture consisting of vaporized fuel and air. As soon as this mixture became sufficiently intimate and its temperature was raised to the proper degree both by the process of compression and by contact with the hot surface (the heated vapor-

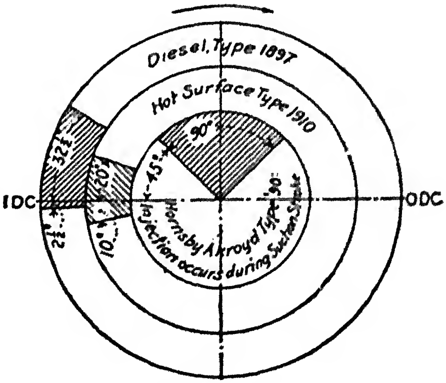


FIG. 2.

izer or other device for imparting heat), ignition was caused and combustion took place. With this system the pressure of compression before ignition, therefore, was thus limited, otherwise pre-ignition would have resulted before the piston reached its inner dead center, causing loss of power and improper operation. Water jacketing of part of the vaporizer chamber in some later designs decreased the temperatures and allowed slightly higher compression pressures with resultant

TABLE I.—HISTORICAL DATA OF VARIOUS ENGINES.
4-CYCLE TYPE.

Date	B. H. P.	Compression Pressure	Injection Period	M. E. P.	Maximum Pressure	Fuel consumption per B.H.P. Hour	Thermal Efficiency	Method of Starting	Grade of Fuel
1888	10	20 Lbs.	Air Inlet	44	120	1.05	12.8	Heating Lamp	Refined
1890	15	40 Lbs.	" "	35	120	1.00	13.5	" "	"
1893	5	40 Lbs.	" "	31.8	..	.99	14.0	" "	"
1898	8	40 Lbs.	" "	43	..	0.73	19.0	" "	"
1899	25	60 Lbs.	" "	40.7	..	0.74	18.0	" "	"
1905	30	60 Lbs.	" "	48	168	0.74	18.0	" "	"
1908	77	180 Lbs.	End of Com'sion	60	..	0.60	22.4	" "	Borneo Crude
1910	51	280 Lbs.	" " "	78	..	0.45	31.0	" "	Russian "
1914	25	420 Lbs.	" " "	80	..	0.48	30.0	Cold "	" "
1917	140	275 Lbs.	" " "	78	445	0.48	29.5	Heating Lamp	Crude
1919	200	375 Lbs.	" " "	78	..	0.48	30.0	Cold	"
1920	120	430 Lbs.	" " "	85	500	0.40	35.0	"	"

higher thermal efficiency. In Table I. data of historical interest regarding oil engines is tabulated.

In 1906 the Diesel patents lapsed and then many makers of hot surface type oil engines adopted one salient feature of that system, namely, the injection of the fuel at the end of the compression stroke (see Fig. 2), and then during the process of compression only air was compressed in the cylinder and pre-ignition could not take place even when the compression was carried so high that the temperature necessary for ignition was exceeded, the fuel being injected into the vaporizer or combustion space at or near the end of compression.

STARTING.—The elimination of the heating lamp for starting purposes in many designs is another great advantage. This has been accomplished by increasing the compression pressure to, or above, 300 — 350 lbs. in the larger size engines. The heat, generated by compression of the air to this pressure is sufficient for cold starting in connection with the more efficient methods of solid or airless injection of the fuel referred to. The cooling effect on the cylinder walls or combustion space of the air blast entering at 800 to 1,000 lbs. pressure and expanding to the pressure existing in the cylinder, as used in the air-blast Diesel engine, is eliminated. With the smaller sized engines (about 50 H.P. in one cylinder and less) some auxiliary heating device such as heating lamp, electric coil, etc., is still used. In the Crossley Oil Engine a small tube heated externally is inserted into the combustion space. (See Chapter V.) In this way the heat

necessary to cause the first few ignitions is furnished, as the temperature generated by comparison alone in the smaller size engines is not sufficient under all working and temperature conditions to insure ignition. This improved method of starting has been found most advantageous in marine and other installations where quick starting is necessary. Its advantage where fire hazard is great has also been proved. The external heating lamp in the older engines frequently required special precautions which are now unnecessary with the modern oil engine.

In England the hot surface type engine is made up to 160 H.P. in one cylinder or 320 H.P. twin cylinder design. In the U. S. A. a four-cylinder 600 B.H.P. four-cycle type with cylinders 21″ diameter and 34½″ stroke is the largest unit built for stationary purposes. For large installations the Diesel Engine is preferred, inasmuch as the skilled attendants necessary with that type can be provided where units of larger earning capacity are employed. The Diesel Engine forms the subject of a separate treatise.*

Ignition.—With practically all modern oil engines the point where ignition begins is therefore simultaneous with the point in the cycle where fuel injection starts. Thus a higher range of pressures including that of compression is found in all modern oil engines as compared with the older types, the clearance in the cylinder and combustion space has been lessened and the volume of the vaporizing chamber and valve box

*Marine and Stationary Diesel Engines by A. H. Goldingham.

has been reduced so as to allow a greater compression pressure being obtained. The spraying or pulverizing devices for injecting the fuel into the combustion space have also received greater attention in later years, with the result that modern engines are equipped with sprayers or pulverizers of greatly improved design (as described in Chapter II), known as the "Airless" or "Solid Injection" type. In some designs high pressure air is injected with the fuel in a similar way to that of the Diesel Engine as described hereinafter, the objective in all systems and designs of sprayers or pulverizers being the thorough breaking up of the particles of fuel as they enter the combustion space of the cylinder, proper distribution and the exposure of the maximum amount of the surface of the globules of fuel or vapor to the air. These two features (*a*) higher compression pressures, (*b*) improved sprayers, may be stated as the chief causes of the increased thermal efficiency of modern oil engines as compared with those of the earlier types. This is plainly indicated in the diagrams shown in Fig. 1, and in Table I, giving particulars of various designs of oil engines constructed between 1890 and 1922. Such improvement follows the laws of thermodynamics* discussed at length in many treatises devoted to that subject briefly outlined as follows†.

If heat is furnished to a perfect heat engine at absolute temperature T and the absolute temperature of

*"Thermodynamics of the Gas, Petrol and Oil Engines," by Sir Dugald Clerk. "Thermodynamics" by Dr. C. E. Lucke. "Steam and Other Heat Engines," J. A. Ewing.

†"Thermodynamics of Steam and Heat Engines," by Peabody.

the source of cold is T', then the efficiency (E) of the engine is

$$E = \frac{T - T'}{T} = 1 - \frac{T'}{T}.$$

THE ISOTHERMAL LINE is that showing the relation between the pressure and volume of a gas due to expansion or compression at constant temperature.

THE ADIABATIC LINE is that denoting the relation between the pressure and volume of a gas due to expansion or compression when no transmission of heat takes place. No heat is extracted or added during the change of volume of the gas.

Where γ = ratio of the specific heat of gas at constant temperature and pressure.

The pressures at different points in the adiabatic curve are related by the equation

$$pv^{\gamma} = \text{constant}.$$

According to Rankine,

$$\gamma = 1.408 \text{ for air.}$$

For hot surface (constant volume) engines

$$\gamma = 1.30 \text{ to } 1.35.$$

A vapor mixture or gas can be heated at constant pressure with the volume varying, the *specific heat* then being the amount of heat necessary to raise the temperature of 1 pound of the gas 1°, the pressure remaining constant; or the gas can be heated at constant volume, the pressure altering, and in that case the *specific heat* is the amount of heat necessary to increase the temperature of 1 pound of the gas 1°, the volume remaining constant.

THE AIR STANDARD diagram first used by Sir Dugald Clerk indicates the actual physical properties of the working fluid but neglects heat losses to cylinder walls. It represents ideal performance, which in the real engine may be approached but never attained. The relative perfection of an engine may be gauged by the closeness with which it approaches this ideal. As the Carnot cycle would represent the ideal efficiency of a heat engine and the Rankine cycle that in steam practice, so the air standard cycle serves the same purpose with the internal combustion engine. This cycle assumes that gases obey the laws of Charles and Boyle within practical limits, that combustion is complete and takes place instantaneously with the piston at the inner dead center and that the gases are always in thermal and chemical equilibrium.

COMPRESSION CYCLES classified by Clerk for engines operating with adiabatic compression and expansion are (*a*) constant temperature, (*b*) constant pressure, (*c*) constant volume. In "a" adiabatic compression raises the temperature through the entire range, the total heat is received during isothermal expansion at the upper temperature. Adiabatic expansion reduces the working fluid from the upper to the lower temperature. The heat discharged is rejected by isothermal compression at the lower temperature. This constitutes the Carnot cycle. In "b," constant pressure, adiabatic compression raises the pressure from the lower to the higher limit, and the heat is added at the upper constant pressure and increasing temperature. Adiabatic expansion reduces the working fluid from

the upper to the lower constant pressure, the heat ‹ charged is rejected at the lower constant pressure ꞓ diminishing temperature. In "c," constant volu: adiabatic compression raises the temperature thro a certain range. Heat is supplied above that range constant volume so that both pressure and temperat then increase. Adiabatic expansion reduces the te perature through a certain range and increases the ume to that existing before compression. Heat ‹ charged is rejected at constant volume and dimin: ing temperature.

Where $t =$ temperature before compression;
$v =$ volume before compression;
$t_c =$ temperature after adiabatic compressi
$v_c =$ volume after adiabatic compression ;
$E =$ thermal efficiency,
$K_v =$ specific heat at constant volume.

$$E = 1 - \frac{t}{t_c},$$

and it can be shown that $\left(\frac{v_c}{v}\right)^{\gamma-1} = \frac{t}{t_c}$,

$$E = 1 - \left(\frac{v_c}{v}\right)^{\gamma-1}$$

and if $\frac{v_c}{v} = \frac{1}{r}$ . . . the compression ratio,

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

THERMAL EFFICIENCY thus depends only on the ratio of the maximum volume before compression to the volume after compression. The constant pressure and constant volume are the best cycles for maximum efficiency and maximum power for given stresses in which sufficient compression followed by effective expansion is an essential feature. Without adiabatic compression such expansion is very limited. Compression before ignition, in the internal combustion engine extends the range of effective expansion.

HEAT LOSSES in internal combustion engines are as follows:

(*a*) Loss through the walls externally cooled when gases are at maximum temperature and during expansion. (*b*) Throttling of air inlet and back pressure during exhaust. (*c*) Incomplete combustion at maximum temperature and loss through exhaust. (*d*) Varying specific heat increasing with rise in temperature.

THE EFFICIENCY of an engine, where H is the total heat taken in by the engine, and H' is the heat discharged after performing work, the portion disappearing in work being $H - H'$ and with no other heat loss, would be

$$E = \frac{H - H'}{H},$$

In Fig. 3 where expansion is such as to allow the heat to be discharged with the volume similar to that exist-

ing before compression, the heat supplied to the cycle is

$$H=K_v(T-t_c),$$

heat discharged is

$$H'=K_v(T'-t),$$

and efficiency

$$E=\frac{K_v T-t_c-K_v(T'-t)}{K_v(T-t_c)},$$

$$E=1-\frac{T'-t}{T-t_c}.$$

Both curves are adiabatic and pass through similar volumes

$$\frac{T'}{T}=\frac{t}{t_c}$$

so that $$\frac{T'-t}{T-t_c}=\frac{T'}{T}=\frac{t}{t_c}$$

The efficiency may therefore be expressed,

$$E=1-\frac{T'}{T} \text{ or } 1-\frac{t}{t_c}$$

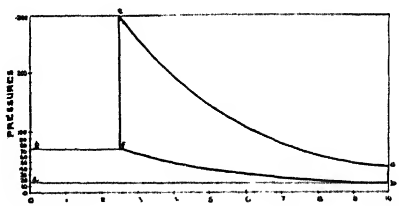


FIG. 3.

Temperatures	*Pressures*	*Volumes*
t absolute temperature at b	p absolute pressures at b	v_o volume at b
t_c absolute temperature at f	p_c absolute pressures at f	v volume at c
T absolute temperature at e	p_o absolute pressures at e	v volume at f
T' absolute temperature at c		

THE CYCLES of operation are also outlined on page 18, Part II. Advantages of the four-cycle are (a) lower fuel consumption, (b) more complete combustion of fuel, (c) simplicity of construction (all parts being accessible or in view with the open crankcase), (d) use of lowest grades of fuel without carbonization. The advantages of the two-cycle are (a) absence of air inlet and exhaust valves and valve motions, (b) power developed of piston displacement per unit of volume as compared with the four-cycle type being 75 to 90 per cent. greater, (c) more even crank effort, (d) lighter flywheel than the four-cycle type. The disadvantages of the four-cycle design as compared with the two-cycle type are: more variable crankpin effort, greater total weight per H.P., necessity of valves and valve motion. Those of the two-cycle are: inferior combustion of fuel and smoky exhaust gases, greater lubrication and more cooling water required, greater fuel consumption, possibility of air leakage when compressed in the crankcase.

VARIOUS EFFICIENCIES.—Thermal efficiency and mechanical efficiency are discussed on pages 86 and 87, Part II. The volumetric efficiency is the ratio between the weight of air contained in the cylinder of a four-cycle engine when the compression stroke begins and

that of the air required to fill the same volume with air at atmospheric pressure. In a two-cycle engine the percentage of pure air present in the total weight of gas or burnt products filling the cylinder at the commencement of compression must also be taken into account.

CHAPTER II

DESIGN AND CONSTRUCTION OF PARTS—VAPORIZERS—SPRAYERS

In the modern hot surface oil engine a higher range of pressures is found as compared with that of those discussed in Part II. In some designs the pressures (as will be observed from the indicator cards reproduced hereinafter) are similar to those of the Diesel type and therefore with increased pressures, all parts are made heavier to withstand these greater stresses than was necessary in the low pressure engines discussed in Part II. Accordingly formulæ necessary for such modern designs are given here, as those stated in Part II would not be applicable with the higher range of pressures.

MAIN FRAME AND CYLINDER.—It will be noted from the various sectional illustrations of the different designs of horizontal frames that in most cases they are now carried further back, so as to fully support the cylinder over its whole length. The design found in the older types with the "over-hung" cylinder has now given place to this construction. In some cases the foundation bolts are brought up through the frame so as to hold it solidly and without vibration on the concrete foundation. The sectional views of the vertical types show the arrangement of the main bearings with

facilities for their adjustment. In the cheaper designs the babbitt of the bearing is inserted directly into the main casting, but in all larger and high-class engines the bearing consists of a separate shell readily removable for repairs or refitting without removing the crankshaft. In all designs it is of the greatest importance that the exact alignment of the crankshaft in its bearings is preserved and the equal wear of all main bearings be assured so that full bearing on all journals and correct alignment is maintained when the engine has become much worn. To equalize the wear on the main bearings the De La Vergne Machine Co. place the flywheel on some of their horizontal engines close up to the outboard bearing instead of putting it near the main frame bearing, thus the weight of the flywheel and the wear on the bearing consequent on its weight is taken solely by the outboard bearing and the wear on the main frame bearing nearest it, is reduced and allows the wear on the two bearings to remain equal and thus exact alignment is preserved.

THE CYLINDER AND LINER are discussed on page 24, Part II. With modern engines the liner is generally cast separately. The tension load is taken through the cylinder casing and the radial stresses are taken by the liner. In some vertical designs the tension stresses are taken solely by long bolts passing from the bed plate to the cylinder head. The liner is made of hard close-grained cast iron, having a tensile strength of about 35,000 pounds per square inch. It is usually inserted into the cylinder casing held rigidly at its back or upper end, supported in the center and held in place by a

rubber ring at its front or lower end; thus the expansion (due to varying temperatures of the inner liner and outer casing when in operation with the cooling water jacket between them) is allowed for. Assuming the combustion pressure to vary from 400 to 600 pounds per square inch, and assuming an average pressure of 510 pounds per square inch of piston area, the pressure on piston (P) would be:

$$P = 510 \times 0.785D^2 = 400D^2,$$

where D = Diameter of Cylinder.

The thickness (S) of cylinder liner may be taken as:

$$S = 0.07\ D \text{ inch.}$$

To this add ¼ inch to thickness of metal as allowance for reboring the liner when it becomes worn. With more than 15 inches diameter the liner thickness of metal may be gradually decreased toward the open end, to 75 per cent. of S.

The cylinder casing or jacket wall has to withstand in the direction of its axis a pulling force $P = 400d^2$. The cross-sectional area $d = \pi\ D\ s$. The stress per square inch

$$P = \frac{400 \times d^2}{\pi\ D\ s} \quad \text{or} \quad s = \frac{400 \times d^2}{\pi\ D\ p}$$

and let p = 1800 pounds per square inch.

$$\text{We get } s = 0.071 \frac{d^2}{D}$$

Where D = Mean diameter of cylinder wall.
s = Thickness of cylinder wall.
d = Diameter of piston.

Cylinder head bolts should be of good wrought iron or soft with an allowable tensile stress of 5500 to 6500 pounds per square inch. Size of bolts is determined by maximum pressure $(Pt) = 400\ D^2$, 20 per cent. to 30 per cent. added for tightening.

COMBUSTION SPACE.—Examination of the various sectional views shows a simpler form of combustion space than in the older types. The air and exhaust valves are brought closer to it, and the passages to the valve box, and all other spaces in the older engines have been eliminated so as to afford less clearance necessary to procure greater compression pressure as referred to more fully in the descriptions hereinafter of each design.

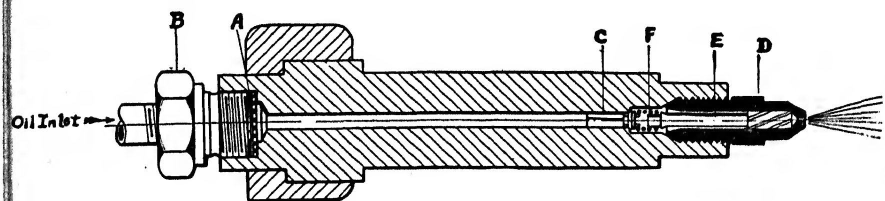


FIG. 4.—A. Copper gauze filter. B. Locknut on oil supply pipe. C. Check valve. D. Spray nozzle. E. Thread holding nozzle in spray valve body. F. Valve spring.

SPRAYERS.—Reference has already been made to the improvements made in the spraying or pulverizing devices of modern oil engines and to the fact that the increased thermal efficiency is due in great measure to the attention given to this part. It is generally a simpler apparatus than the pulverizers necessary for Diesel

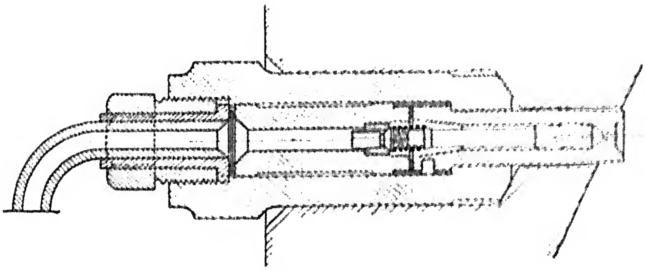


FIG. 5.

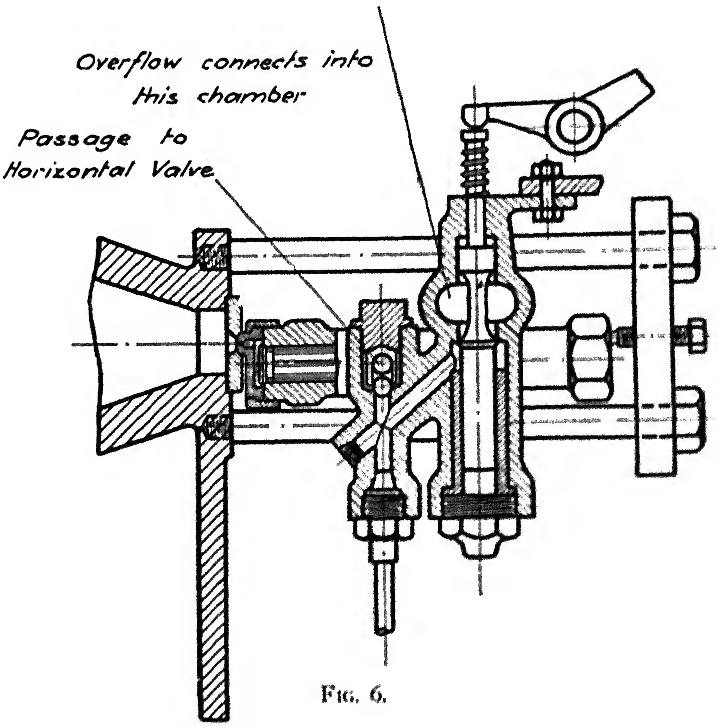

Overflow connects into
this chamber
Passage to
Horizontal Valve

FIG. 6.

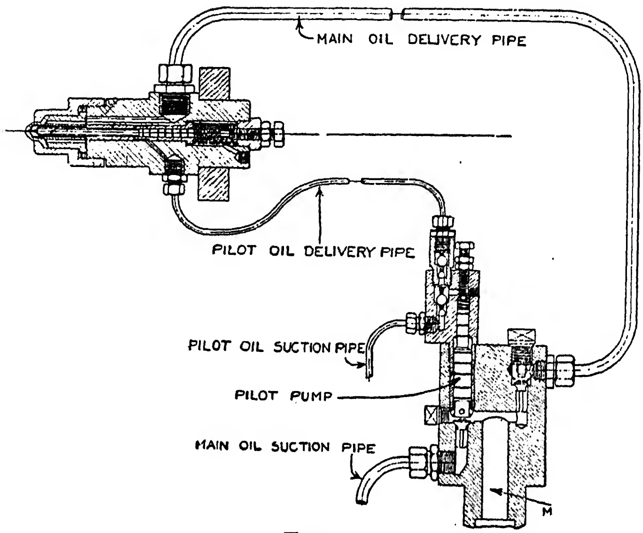


FIG. 7.

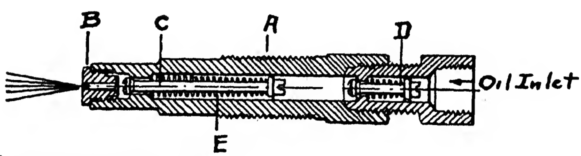


FIG. 8.—A. Spray valve body. B. Spray nozzle. C. Inner check valve. D. Outer check valve. E. Valve spring.

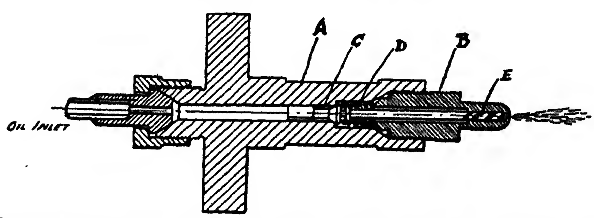


FIG. 9.—A. Spray valve body. B. Spray nozzle. C. Check valve. D. Spring. E. Spirals on valve stem periphery.

engines. The function of the sprayer is to vaporize or break up the particles or globules of the liquid fuel and distribute the vaporized spray or fog throughout the whole space of the combustion chamber or vaporizer so that the globules have the maximum amount of their surface exposed to the oxygen of the air into which they are sprayed. The small steel nozzle has a varying dimension from .020 to .060 inch. With the larger hot surface type oil engines and those having air blast injection similar sprayers or pulverizers are used as with Diesel engines.

The fuel sprayer as used on the De La Vergne "FH" type where the air blast enters the combustion space with the fuel is shown in Fig. 7-a, Part II. The De La Vergne "DH" sprayer is shown in Fig. 4, and the sprayer employed by that company on their later design "SI" (solid injection) oil engine is illustrated at Fig. 5. The method of spraying in this design is described hereinafter. The Hornsby type "R" sprayer is shown at Fig. 6, and the Ruston sprayer used when light and heavy fuels are burnt and where a "pilot" ignition requiring about 5 per cent of lighter fuel to be injected first and followed by the heavy fuel is shown in Fig. 7. The sprayers used in two-cycle engines are shown in Figs. 8 to Fig. 10.

REGULATION OF SPEED is effected by means of a pendulum governor arranged to vary the amount of fuel entering the vaporizer or combustion space in accordance with the load carried and to maintain an even speed of rotation of the crankshaft within the following limits: With fairly even load the speed variation

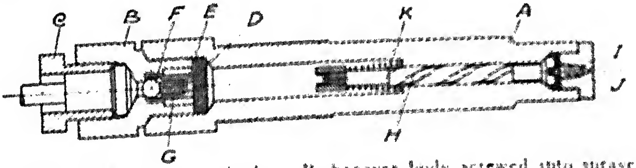


Fig. 10.—A. Sprayer cylinder. B. Sprayer body screwed into sprayer cylinder. C. Inlet oil nipple. D. Filter. E. Sleeve. F. Ball check valve. G. Valve spring. H. Spiral grooves in check valve. I. Fuel inlet spray. J. Nozzle. K. Spring maintaining valve in position.

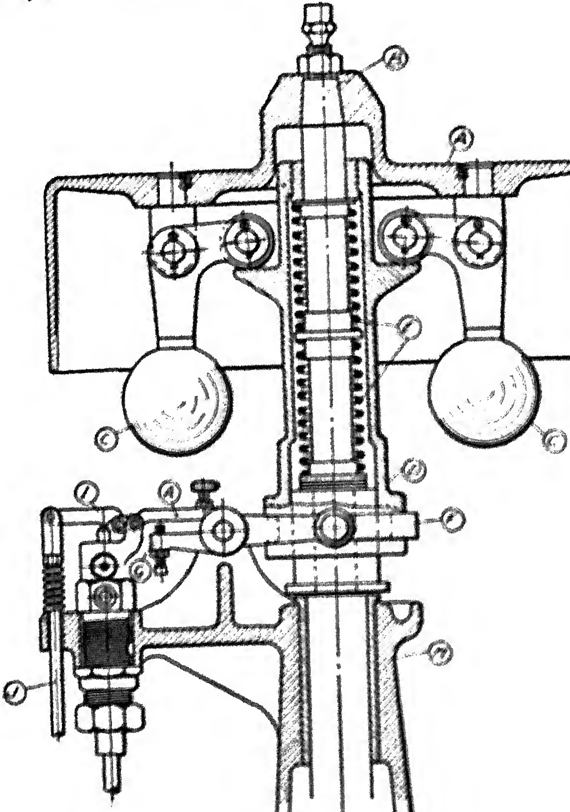
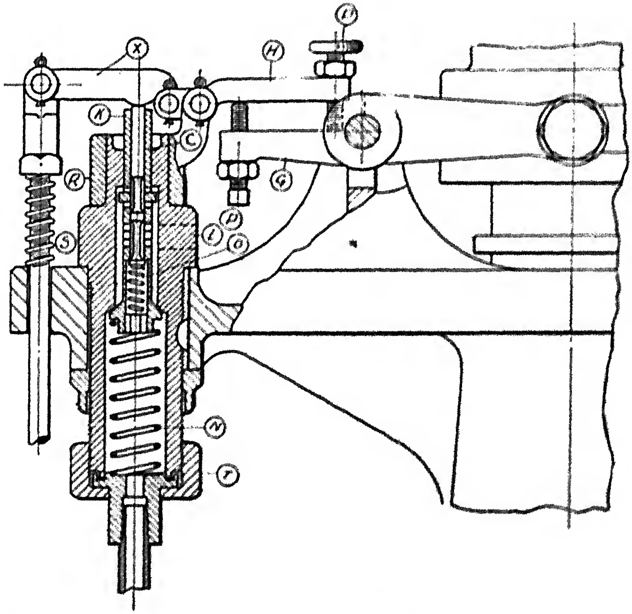


FIG. 11.

should not exceed 1¼ per cent each side of the uniform speed of rotation; when the load is varied from full load to ¼ load the speed variation should not exceed 4 per cent and from full load to no load not over 5 per



Figs. 11 and 12.—A. Governor head. B. Governor spindle. C. Governor balls. D. Sleeve. E. Springs—one wound right hand and one left. F. Yoke pivotted on E. G. Fulcrum lever. H. Valve lever. J. Connection to oscillator actuated by cam. K. Small over-flow valve. L. Entrance for fuel from pump. N. Spring closing smaller valve K. O. Spring closing larger over-flow valve. P. Adjusting screws. R. Cap for supporting fulcrum. S. Valve casing. T. Overflow to reservoir. X. Lever in contact and moved by screws P.

cent. There are two methods of operation. The governor either (*a*) by-passes the fuel not required through an overflow valve so that the correct amount required enters the vaporizer as shown in Fig. 11 and Fig. 12, or (*b*) lengthens or shortens the stroke of the pump furnishing the fuel as shown in Fig. 13.

The oil pump, as shown in Fig. 14, is employed to raise the liquid fuel, after it has been properly strained of all impurities, to the sprayer. As high pressures are frequently developed in the connection between the oil pump and the sprayer, the pump body and other parts are heavily designed and with a high factor of safety. Two suction and discharge valves are placed in the pump so as to insure proper functioning in continuous operation should grit or dirt be carried to them in the fuel. The pump of modern design (Fig. 14) has no packing and is known as the "packingless" type. It is employed with the Hornsby, the De La Vergne, the Bolinder engines and by other makers. The plunger is made an exact fit to the pump body and is grooved as shown.

Connecting Rods are also discussed on page 32, Part II. A modern design is that shown at Fig. 15 composed of mild steel with adjustable bearings at both ends, that at the piston end so arranged that the adjustment is made before being put in place. This system has proved advantageous inasmuch as it prevents improper adjustment often occurring where the wedge and screw were employed in earlier engines. The hole shown drilled through its center allows efficient lubrication, the lubricant thus passing from the crankpin

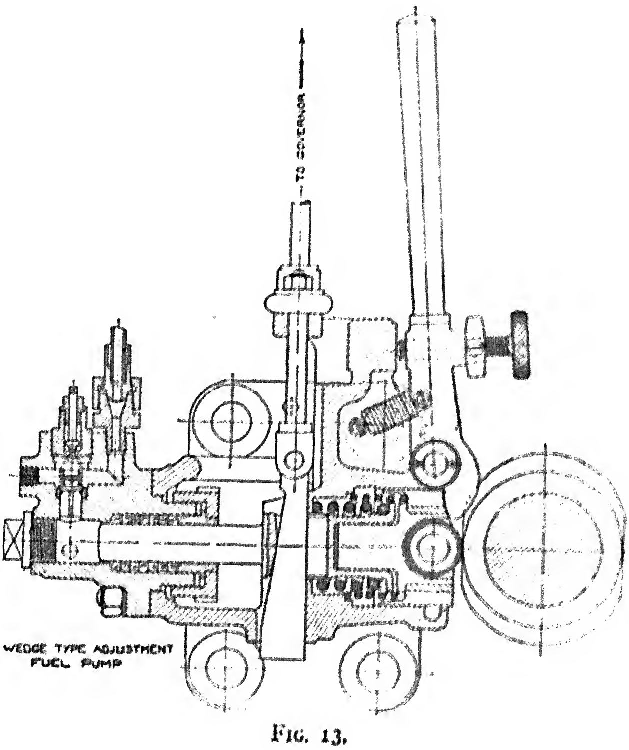


FIG. 13.

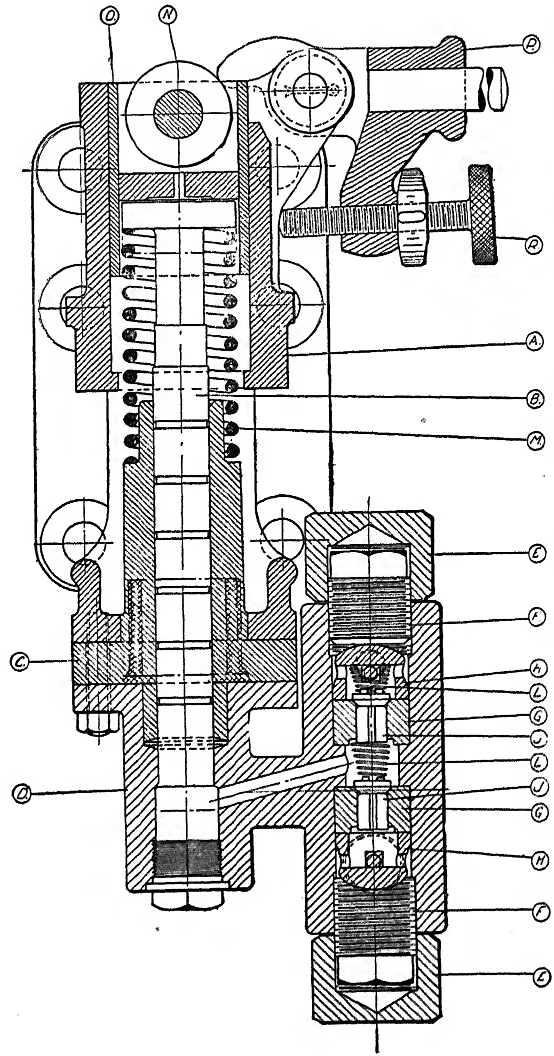


FIG. 14.—A. Pump bracket. B. Pump plunger ground and lapped to fit C. C. Plunger guide of sufficient length to prevent leakage. D. Pump body bolted to A. E. Caps locking F in position. F. Screw plugs holding valve covers in place. G. Valve seats. H. Covers holding valves in place. J. Valves on suction and discharge. L. Springs above suction and discharge valves. M. Spring maintaining contact of roller and plunger. N. Cam roller always in contact with plunger. O. Plunger head. R. Attachment for pumping by hand.

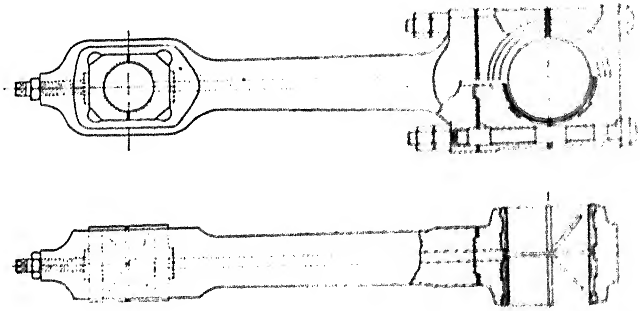


FIG. 15.

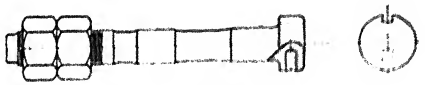


FIG. 16.

to the piston end bearing. A satisfactory formula is as follows:

$$d = .0164 \sqrt[4]{m\ P\ l^2}$$

Where P = load on piston
d = Diameter of rod
m = 30 with 200 ft. per minute piston speed
= 20 with 400 " " " " "
= 15 with 600 " " " " "
= 10 with 800 " " " " "
l = Connecting Rod centers in inches.

THE CONNECTING ROD BOLT shown in Fig. 16 should be made of the toughest wrought iron. The cross section at the bottom of the threads of the bolts should be such in a four-cycle type that the stress as the suction

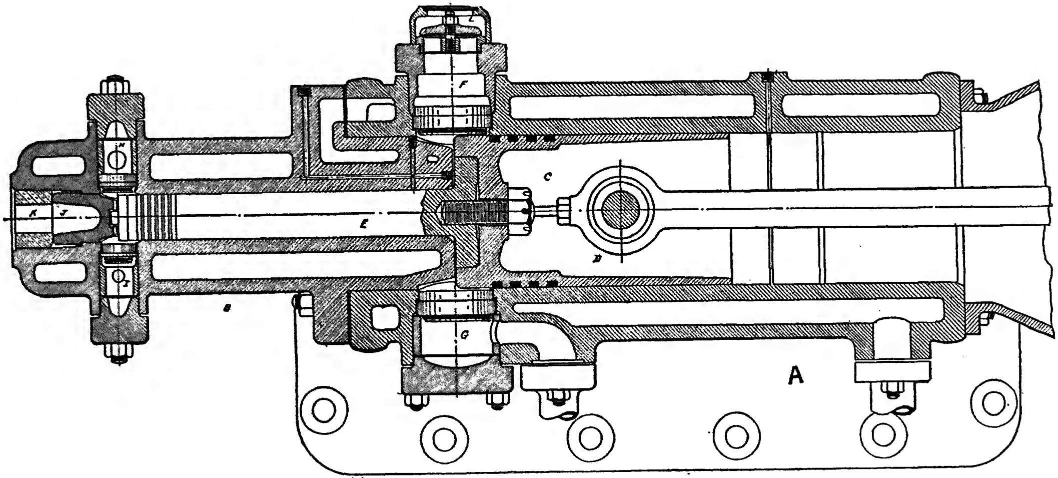


FIG. 17.—A. Water jacketted low pressure cylinder and bracket. B. High pressure cylinder water jacketted. C. Low pressure piston. D. Low pressure wristpin. E. High pressure piston securely bolted to C. F. Low pressure suction valve. G. Low pressure discharge valve. H. High pressure suction valve. J. High pressure discharge valve. K. J. High pressure cylinder head.

stroke commences does not exceed 6000 lbs. per sq. in.

AIR COMPRESSORS.—With some hot surface oil engines air injection with the fuel is found similar to the Diesel air blast injection type. Fig. 17 shows the two-stage compressor as employed on De La Vergne "FH" oil engine. The displacement of the low pressure stage with four-cycle single acting engines should exceed 0.3 cu. ft. per minute per B.H.P. or 18 cu. ft. per B.H.P. hour. The valves used on this compressor are shown at Fig. 18. The area of the air inlet valve may be approximately one-tenth the area of the piston and that of discharge valves one-seventh the area of the piston. To avoid carbonization or other trouble ample cooling surfaces and intercooler surface is provided for.

VALVES.—The inlet valve is designed for a mean velocity of 120 to 140 feet per second.

$$a = \frac{V}{v} \text{ sq. ft.}$$

or

$$a = \frac{F \times c}{v} \text{ sq. ft.}$$

where V = displacement of cylinder in cubic feet per second;
v = mean velocity in feet per second;
a = area of valve in square feet;
c = piston speed in feet per second;
F = area of piston in square feet.

The exhaust valve is usually made of the same size as the air inlet valve, different designs of which are

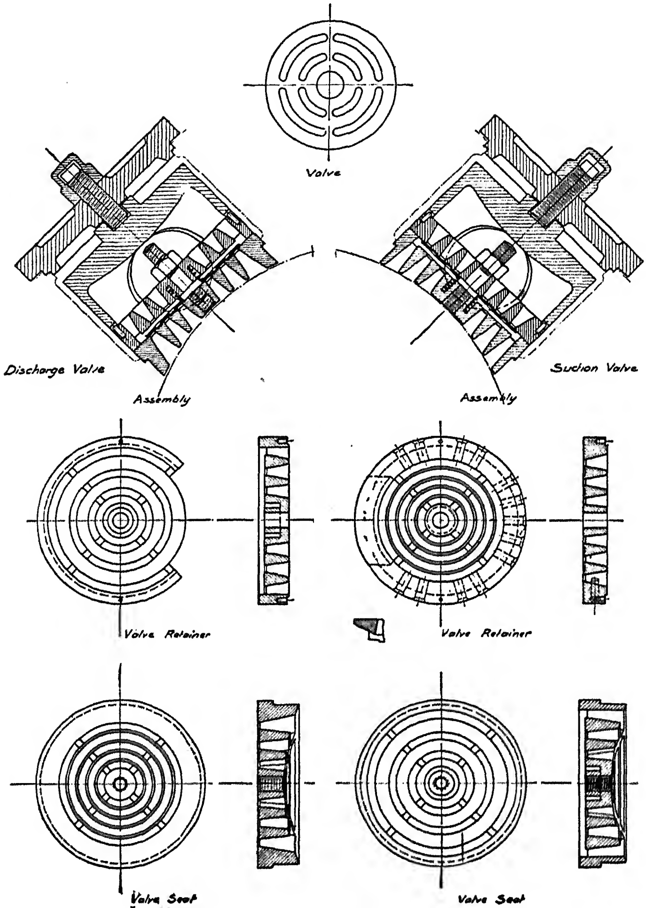

Valve
Discharge Valve
Assembly
Suction Valve
Assembly
Valve Retainer
Valve Retainer
Valve Seat
Valve Seat

FIG. 18.

shown at Fig. 19 and page 42, Part II, as well as in the sectional views of the various engines described in Chapter V.

Exhaust ports with two-cycle engines generally oc-

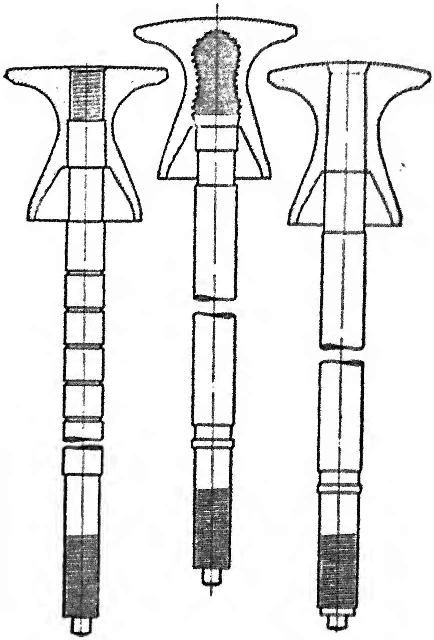


FIG. 19.

cupy half the circumference of the cylinder and are of such height as to allow proper opening and closing by the piston.

PISTONS.—The trunk piston is also discussed on

page 33, Part II. Pistons of later design for use in high compression hot surface and Diesel engines are shown in Fig. 20. They are constructed in two parts bolted together with ground joint. The dimensions of the various parts as shown in Fig. 20 are:

D_1 Diameter of wristpin 0.3—0.4 D
L_1 Length of wristpin bearing 0.5 D
L_2 Length wristpin bearing in piston 0.25 D
D_2 Diameter of wristpin boss 0.2 D
T_1 Thickness head of piston 0.08—0.125 D.

Thickness of metal must be sufficient to insure proper cross section and to allow for radiation whether ribs reinforce it or not.

T_1 Thickness of piston barrel head end 0.09—0.1 D.
T_2 Thickness of piston barrel crank end 0.06—0.75 D.
Number of piston rings 6—7.
Width of piston rings 3/8"—1/2".
Depth of piston ring grooves 3/8"—3/4".
Clearances for expansion as follows:
Head end to rear of last piston ring 2½/1000 per inch.
Diameter tapering 1½/1000 per inch. Diameter of body.

CRANKSHAFT.—The discussion of crankshafts on page 26, Part II, is applicable to the earlier low compression engine. For the high pressure hot surface engines of many designs the stresses obtaining are very similar to those of the full Diesel engine while in other designs a slightly lower range of pressures is found.

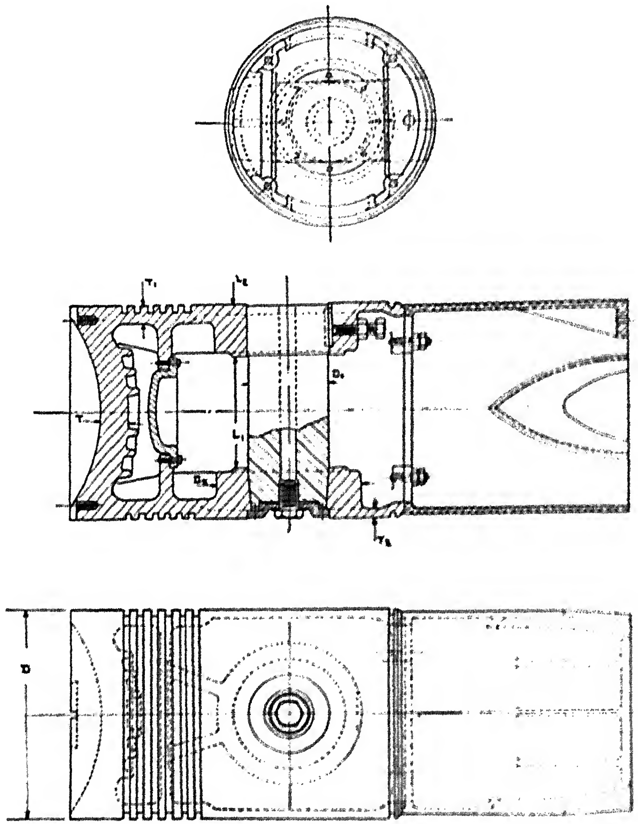


FIG. 20.

The following data is applicable therefore for these engines as well as for those of the Diesel stationary type. The calculations governing the design of crankshafts discussed in many treatises on the dynamics of reciprocating engines, take into consideration first, the load on the piston shown by the indicator card, second, the stroke and maximum speed of engine, the acceler-

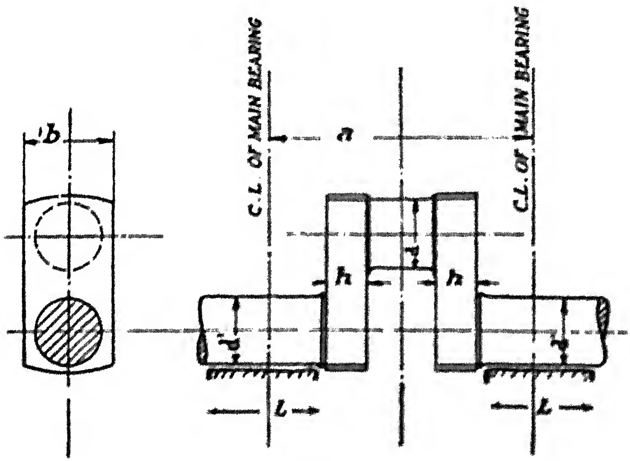


FIG. 21.

ation of the piston, and the inertia forces resulting from it and other reciprocating parts, as well as the length of the connecting rod. The twisting moment on the crankshaft due to the stresses of the above forces allows the maximum twisting moment to which the shaft will be subjected, to be obtained, and the dimensions of the crankshaft to be computed accordingly.

The following are dimensions of representative high pressure hot surface or stationary Diesel engine shafts with average pressures: (See Fig. 21.)

D = Diameter of piston.

a = 2.2 D. d′ = 0.55 D. d = 0.60 D.

$$l = \frac{D}{1.65} \quad L = 0.85\ D. \quad h = 0.6\ d.$$

b = 1¼ d to 1 1/3 d.

The dimensions "h" and "b" assume that an outboard bearing is used.

Table II gives actual dimensions taken from various crankshafts of low compression, high compression, and Diesel oil engines in successful operation.

Some engine builders are now employing built-up crankshafts instead of those forged in one piece. In Table III are given dimensions of low compression, high compression and Diesel engine crankshafts.

TABLE II.—DIMENSIONS OF REPRESENTATIVE LOW, HIGH COMPRESSION AND DIESEL CRANKSHAFTS.

Type Engine	Low Compr.	Low Compr.	Low Comp.		High Compression or Diesel		
B.H.P. (One cyl.)	25	40	65	50	100	140	180
Type Shaft *	Forged	Forged	Either	Forged	Either	Either	Either
Diam. Crankpin	4"	5½"	6½"	7½"	8"	10"	13"
Length Crankpin	4⅛"	5¼"	7½"	7"	8½"	11"	10"
Diam. Main Bearing	4"	4¾"	6"	6¾"	8¼"	10"	11"
Length Main Bearing	9"	11½"	13¼"	9½"	14½"	17½"	18¼"
Diam. Flywheel section	4"	5¾"	7"	7½"	9½"	11⅝"	11⅝"
Diam. Outb'd bearing		3½"	5"	6"	7"	8"	9"
Length Outb'd bearing		7"	8"	8"	10"	11"	15"
Thickness Crank Web							
" Forged:	2¾"	3¾"	4¾"	3⅜"	5½"	6¾"	6¾"
" Built-up:			5"		5⅝"	7"	7⅝"
Width Crank Web							
" Forged:	5"	7¼"	8"	15 diam. Circular	12½"	13"	15"
" Built-up:			12"		18"		26"
Diam. Coupling End	4"	3½"	6"		7"	8"	9"

TABLE III.—CRANKSHAFT PROPORTIONS

Type Engine	Forged pression	Forged	Built up pression	Built up
Type Shaft	Forged	Forged	Built up	Built up
D—Diam. in Crankpin	D	D	D	D
D1—Diam in Main Bearing	.9 D	.85D	.9 D	.85D
D2—Diam. in Flywheel Section	1.1 D	.9D	1.1 D	.9 D
D3—Diam. in Outboard bearing	.75D	.7D	.75D	. 7D
L—Length Crankpin	1.15D	1.00D	1.15D	.85D
L1—Length Main Bearing	2 D	1.4D	2 D	1.4 D
L2—Outboard Bearing	1.2 D	1.15D	1.2 D	1.15D
T—Thickness Crank-web	.73D	.55D	.73D	.6 D
S—Width Crank-web	1.2 D	1.15D	1.8 D	2.0 D

The Diesel shaft has smaller proportional values for parts based on crankpin diameter as compared with those of the low compression engines. This is due to the high mean bearing pressures on Diesel type crankpins requiring a much larger pin than the low compression engines.

The specifications under which various builders require the manufacture of their engine crankshafts for oil engines are shown in Table IV.

TABLE IV.—STEEL FOR INTERNAL COMBUSTION ENGINE CRANKSHAFTS

Use	Low compr. Engines	Diesel	Diesel
Spec.	USA Manf.	USA Manf.	High Speed Engs.
Tensile St.	70,000	80,000	105,000
Elast. Limit	35,000	48,000	80,000
Elong. 2″	22%	25%	20%
Red. Aver.	45%	50%	50%
Carbon	.30—.40	.47—.53	.28—.45
Sulphur	.04 Max.	.04 Max.	.05 Max.
Phos.	.04 Max.	.04 Max.	.05 Max.
Mang.			.48 to .80
Chrom.			.75 to 1.0
Van.			.14 to .16
Treatment	Anneal.	Oil quench heat treat.	Heat treat.

LLOYD'S RULES require the following dimensions for Diesel Engine crankshafts. Where the maximum pressure in the cylinder does not exceed 500 pounds per square inch the diameters of the crankshaft are not to be less than those given by the following formula:

$$\text{Diameter of crankshaft} = \sqrt[3]{D^2 \times (AS + BL)},$$

where D = diameter of cylinder;

S = length of stroke;

L = span of bearings adjacent to a crank measured from inner edge to inner edge.

The values of $(AS+BL)$ are as follows:

4-cycle S. A. engine	2-cycle S. A. engine	Values of coefficient
4 or 6 cylinder	2 or 3 cylinder	$.089S+.056L$
8 cylinders	4 cylinders	$.099S+.054L$
10 or 12 cylinders	5 or 6 cylinders	$.111S+.052L$
16 cylinders	8 cylinders	$.131S+.050L$

For the auxiliary Diesel engines, diameters may be 5 per cent less than above. In solid forged shafts the breadth of the webs should not be less than 1.33 times and the thickness not less than 0.56 times the diameter of the shaft as found above, or, if these proportions are departed from, then the webs must be of equivalent strength. Where no flywheel is employed diameter of intermediate shaft must not be less than that given by this formula.

$$\text{Diameter of intermediate shaft} = \text{coefficient} \sqrt[3]{D^2 \times S},$$

where D = diameter of cylinder;

S = stroke of piston.

The value of the coefficient is given below:

4-cycle S. A. engine	2-cycle S. A. engine	Value of co-efficient
4 cylinders	2 cylinders	.456
6, 8, 10 or 12 cylinders	3, 4, 5, or 6 cylinders	.436
16 cylinders	8 cylinders	.466

Where the stroke is between 1.2 times to 1.6 times the diameter of the cylinder $(.735D + .273S)$ may be substituted for $\sqrt[3]{D^2 \times S}$. If the maximum pressure in the cylinders exceeds 500 pounds per square inch the diameters of shafting throughout must be increased in the proportion:

$$\sqrt[3]{\frac{\text{Maximum pressure in pounds per square inch}}{500}}$$

Balancing of the reciprocating and rotating parts recommended by Haeder & Huskisson is as follows, taking account of the inertia:
For horizontal engines:

$$W_1 = 0.7(W_2 + W_3)\frac{r}{R} \text{ pounds,}$$

and for vertical engines:

$$W_1 = W_2\frac{r}{R} \text{ pounds,}$$

where W_1 = Weight of the balance weight in pounds;
R = Radius of the center of gravity of the balance weight in feet;
W_2 = Weight of the crank pin and the big end of the connecting rod + half the weight of the body of the connecting rod in pounds;
r = Throw or radius of the crank in feet;
W_3 = Weight of the piston and piston pin in pounds + half the body of the rod and small end.

The stroke ratio varies in different designs. In high-speed engines the stroke is 1 to 1.3 diameter, while in slow-speed engines it is 1.3 to 1.6 diameter.

Piston speed should not exceed 900 feet per minute. With greater speed efficient lubrication is difficult and the wear of the cylinder liner may be greater, necessitating frequent reboring or renewals.

Flywheels.—The design and construction of flywheels is discussed on page 35, Part II and formulae

for the calculation of weights, etc., necessary is there given.

THE PARALLEL OPERATION of alternators either belted or direct connected to internal combustion engines is a subject that has been widely discussed everywhere.[1] The requirements of manufacturers of electrical apparatus so as to ensure proper paralleling are that the total flywheel effect should be such that the angular deviation from the uniform speed of rotation within the cycle does not exceed 3½ electrical degrees. A mechanical degree is equivalent to an electrical degree divided by half the number of poles of the alternator. For instance, with an alternator having 48 poles, 6 electrical degrees would be equivalent to 0.25 mechanical degree.

Everest[2] gives the permissible limit of speed irregularity, thus:

$$\text{Degree of irregularity} = \frac{K}{6 \times (\text{number of poles})},$$

where K = Number of impulses per revolution.

The flywheel effect to avoid resonance in foot-tons of stored energy at normal speed per one K. W. equals

$$\frac{1.3 \times \text{poles} \times (\text{strokes per engine cycle})^2}{\text{R.P.M.}}$$

[1]Parallel Operation of Alternators, Inst. E. E., 1908, Dr. E. Rosenberg; Design of Flywheels, etc, by R. E. Doherty and R. F. Franklin, A.S.M.E., 1920.

[2]Journal Inst. E. E. (London) p. 530, 1912.

CHAPTER III

REMARKS ON TESTING—INSTALLATION—OPERATION AND CORRECTION OF OIL ENGINES

THE methods of testing Oil Engines is fully discussed in Chapter III, Part II. Extracts from the code of tests of the American Society of Mechanical Engineers are as follows:

OBJECT AND PREPARATIONS.—Determine the object, take the dimensions, note the physical condition of the engine and its appurtenances, install the testing appliances, etc., and make preparations for the test accordingly.

OPERATING CONDITIONS.—Determine what the operation conditions should be to conform to the object in view, and see that they prevail throughout the trial.

DURATION.—The test of a gas or oil engine with substantially constant load should be continued for such time as may be necessary to obtain a number of successive records covering periods of half an hour or less during which the results are found to be uniform. In such cases a duration of three to five hours is sufficient for all practical purposes.

STARTING AND STOPPING.—The engine having been set to work under the prescribed conditions, the test is begun at a certain predetermined time by commencing to weigh the oil, or measure the gas, as the case may be, and take other data concerned; after which the regular measurements and observations are carried forward until the end. When the stopping time arrives the test is closed by simply taking the final readings.

RECORDS.—The general data should be taken and recorded in the same manner as that described hereafter.

CALORIFIC TESTS AND ANALYSES.—The quality of the oil or gas should be determined by calorific tests and analyses made on representative samples.

HEAT CONSUMPTION.—The number of heat units consumed by the engine is found by multiplying the heat units per pound of oil or per cubic foot of gas (higher value), as determined by calorimeter test, by the total weight of oil in pound or volume of dry gas in cubic feet consumed.

HORSEPOWER AND EFFICIENCY.—The indicated horsepower, brake horsepower, and efficiency are computed by the same methods as those explained on page 23, Part II.

HEAT BALANCE.—The various quantities showing the distribution of heat in the heat balance are computed in the following manner:

The heat converted into work per I.H.P.-hour. (2546.6 B.t.u.) is found by dividing the work representing 1 H.P., or 1,980,000 foot-pounds per hour by

the number of foot-pounds representing 1 B.t.u., or 777.5.

The heat rejected in the cooling water is obtained by multiplying the weight of water supplied by the number of degrees rise of temperature, and dividing the product by the indicated horsepower.

The heat rejected in the dry exhaust gases per I.H.P.-hour is found by multiplying the weight of these gases per I.H.P.-hour by the sensible heat of the gas reckoned from the temperature of the air in the room and by its specific heat. The weight of the dry exhaust gases per I.H.P.-hour is the product of the weight of fuel per I.H.P.-hour by the weight of the dry gases per pound of fuel. The latter is the product of the proportion of carbon in 1 pound of fuel by the weight of the dry gases per pound of carbon, which may be found by the formula

$$\frac{11\,CO_2+8\,O+7(CO+N)}{3(CO_2+CO)},$$

in which CO_2, O, CO, and N are percentages of the dry exhaust gases by volume.

When the weight of air supplied per pound of fuel is determined the weight of dry gas per pound of fuel may be found by the formula

$$1+\text{pound air per pound fuel}-9H,$$

in which H is the proportion of hydrogen in 1 pound of fuel.

The heat lost in the moisture formed by the burning of hydrogen in the fuel gas is found by multiplying the total heat of 1 pound of superheated steam at the temperature of the exhaust gases, reckoning from the temperature of the air in the room, by the proportion of the hydrogen in the fuel as determined from the analysis, and multiplying the result by 9.

The heat lost in superheating the moisture contained in the gas and air is determined by multiplying the difference between the temperature of the exhaust gases and that of the gas and air by the average specific heat of superheated steam for the range of temperature and pressure.

The heat lost through incomplete combustion is obtained by analyzing the exhaust gases and computing the heat of the unburned products which would have been produced by their combustion.

The above rules do not apply to engines with hit-and-miss governors.

DATA AND RESULTS.—The data and results should be reported in accordance with the form given herewith (Table V.), adding lines for data not provided for, or omitting those not required, as may conform to the object in view. If a shorter form is desired, items given in fine print and designated by letters of the alphabet may be omitted. Unless otherwise indicated, the items should be the averages of the data.

TABLE V

DATA AND RESULTS OF GAS OR OIL ENGINE TEST

(1) Test of ....engine, located at..................
To determine ....
Test conducted by ....

DIMENSIONS, ETC.

(2) Type of engine, whether oil or gas ....
(3) Class of engine, (mill, marine, motor for vehicle, pumping or other) ....
(a) Number of strokes of piston for one cycle, and class of cycle ....
(b) Method of ignition..
(c) Single or double acting.......................................
(d) Arrangement of cylinders ....
(e) Vertical or horizontal.......................................
(4) Rated power ..................................H.P.
(a) Name of builder..
(5) Number and diameter of working cylinders ....in.
(a) Number and diameter of compression cylinders........in.
(b) Diameter of piston rods..................................in.
(6) Stroke of pistons ....ft.
(a) Compression space referred to piston displacement.per cent
(b) Stroke of compression piston..........................ft.
(c) H.P. constant for 1 pound m.e.p. and 1R.P.M......H.P.

DATE, DURATION, ETC.

(7) Date ....
(8) Duration ....hr.
(9) Kind of oil or gas ....
(a) Physical properties of oil (specific gravity, burning point, flashing point) ....

TOTAL QUANTITIES

(10) Gas or oil consumed .... cu. ft. lb.
(11) Moisture in gas, in per cent by weight, referred to dry gas ....per cent.
(12) Equivalent dry gas at 60 deg. and 30 in ....cu ft.
(a) Air supplied ....cu. ft.
(13) Cooling water supplied to jackets ....lb.
(a) Water or steam fed to cylinder.....................lb.
(14) Calorific value of oil per lb., or of dry gas per cu. ft. at 60 deg. and 30 in. by calorimeter test (higher value) ....B.t.u.

Hourly Quantities

(15) Gas or oil consumed per hour .... cu. ft. lb.
(16) Equivalent dry gas per hour at 60 deg. and 30 in..cu. ft.
(17) Cooling water supplied per hour ....lb.
(18) Heat units consumed per hour (Item 16 × Item 18) ....B.t.u.

Analysis of Oil

(19) Carbon (C) ....per cent
(20) Hydrogen (H) ....per cent
(21) Oxygen (O) ....per cent
(22) Sulphur (S) ....per cent
(a) Moisture ....per cent
(b) Result of fractional distillations..........................per cent

Analysis of Exhaust Gases by Volume

(23) Carbon dioxide (CO_2) ....per cent
(24) Carbon monoxide (CO) ....per cent
(25) Oxygen (O) ....per cent
(26) Nitrogen (N) ....per cent

Indicator Diagrams

(27) Pressure above atmosphere ....pounds per square inch
(a) Maximum pressure ....pounds per square inch
(b) Pressure at beginning of stroke....pounds per square inch
(c) Pressure at end of expansion.....pounds per square inch
(d) Exhaust pressure at lowest point...pounds per square inch
(28) Mean effective pressure ...... pounds per square inch

Speed

(29) Revolutions per minute .... R.P.M.
(30) Average number of explosions or firing strokes per minute ....
(a) Variation of speed between no load and full load..R.P.M.
(b) Momentary fluctuation of speed on suddenly changing from full load to half load..................R.P.M.

Power

(31) Indicated horsepower ....i.h.p.
(32) Brake horsepower ....br. h.p.
(33) Friction horsepower by difference (Item 39—Item 40)[1] .... ft. h.p.
(a) Friction horsepower by friction diagrams..........fr. h.p.
(34) Percentage of indicated horsepower lost in friction Item 41 ....per cent

[1] In two-cycle engines this includes the power required for compression.

Economy Results

(35) Heat units consumed by engine per i.h.p. per hour[2] ....B.t.u.
(36) Heat units consumed by engine per br-h.p ......B.t.u.
(37) Dry gas at 60° and 30 inches consumed per i.h.p-hr. ....pounds, cubic feet
(38) Pounds of oil or cubic feet of dry gas per br-h.p-hr. ....pounds, cubic feet

Efficiency

(39) Thermal efficiency referred to indicated horse-power ....per cent
(40) Thermal efficiency referred to brake horse-power ....per cent

Work Done per Heat Unit

(41) Net work per B.t.u. consumed (1,980,000÷Item 40) ....foot-pounds

Heat Balance

(42) Heat balance, based on B.t.u. per i.h.p. per hour ....

	B.t.u.	Per cent
(a) Heat converted into work.........	2546.5	....
(b) Heat rejected in cooling water.....	....	....
(c) Heat rejected in the dry exhaust gases	....	....
(d) Heat lost due to moisture formed by burning of hydrogen............	....	....
(e) Heat lost in superheating moisture in gas and air	....	....
(f) Heat lost by incomplete combustion	....	....
(g) Heat unaccounted for, including radiation	....	....
(h) Total heat consumed per i.h.p-hr., same as Item 38..............	....	....

Sample Diagrams

(43) Sample indicator diagrams from each cylinder and if possible a stop-motion light-spring diagram showing inlet and exhaust pressures. ....

[2]If these results, in the case of a gas engine, are based on the low value of the heat of combustion that fact should be so stated.

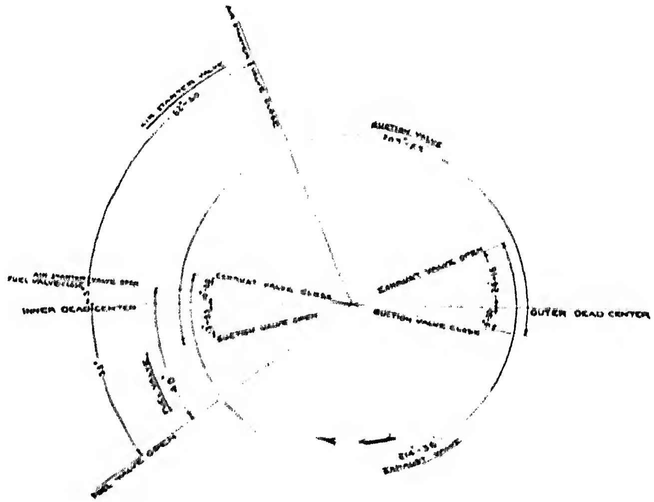


FIG. 22.

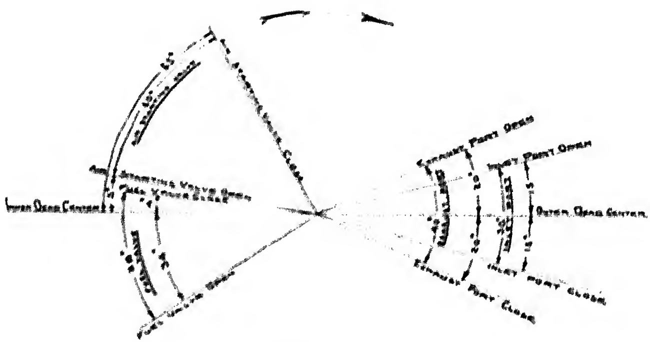


FIG. 23.

VALVE SETTINGS.—The correct setting of the air inlet, exhaust valves, fuel injection period and starting valve opening varies in different designs and each builder usually issues instructions with the engine in this respect. A representative diagram showing the valve settings of a four-cycle hot surface oil engine are shown at Fig. 22, and those of a two-cycle type are shown in Fig. 23. When a four-cycle engine has the direction of rotation of its crankshaft reversed as is required in larger marine installations the sequence of

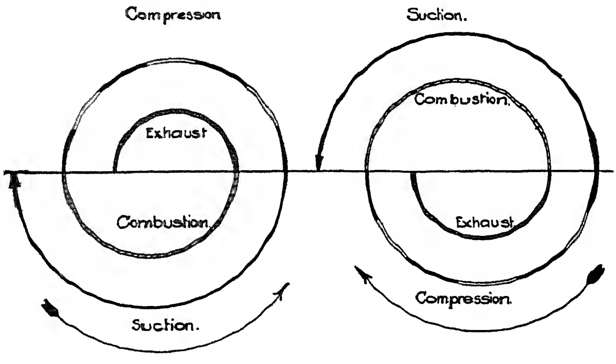


FIG. 24.

valve movements is shown in each direction at Fig. 24.

INSTALLATION.—The complete installation of a modern horizontal hot surface oil engine is shown in Fig. 25. The engine room correctly designed and arranged allows ample space around each engine both for proper attendance and for the removal of any part of the engine that may require inspection or repairs. With

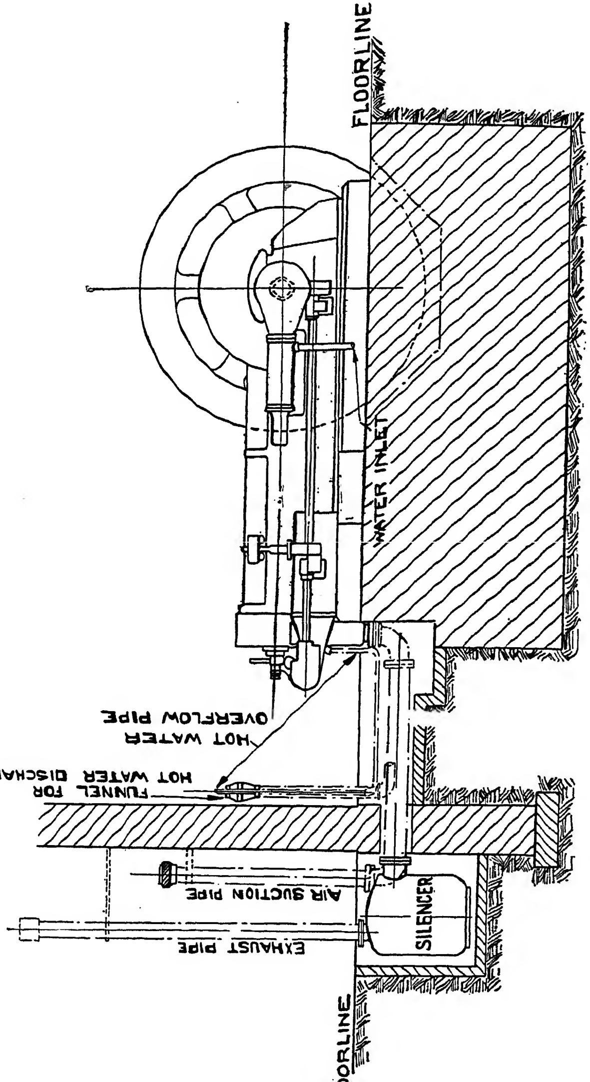

FLOORLINE
WATER INLET
OVERFLOW PIPE
HOT WATER
FUNNEL FOR
HOT WATER DISCHARGE
AIR SUCTION PIPE
EXHAUST PIPE
SILENCER

Fig. 25.

the vertical type sufficient head room should be provided for withdrawing the piston and connecting rod from above, with those designs where this is necessary. A traveling crane overhead is advantageous both during installation and when necessity for repair arises. The engine room should be well lighted and ventilated, and free from injurious gases, the floor should be of dust-proof material and the engine room should be properly heated especially in colder climates so as to prevent freezing in the circulating water pipes or passages of the engine, etc.

THE FOUNDATION for the engine is built of the best concrete, a mixture for which can be composed of one part Portland cement and six to seven parts of sand and broken stone. For horizontal engines the weight of foundation should not be less than 2,000 pounds per B.H.P. for single cylinder types and slightly less per B.H.P. for twin cylinder engines. With vertical and multi-cylinder engines this weight can be decreased slightly. The depth of the foundation should not be less than 5 D when D equals diameter of cylinder. The above dimensions are for use when the foundation is built on solid ground. Where wet ground or quicksand, etc., is encountered piling or other special arrangements must be made to make the ground the equivalent of solid ground before the building of the foundation is commenced. Usually the engine foundation is thoroughly insulated from the foundation of the building or other foundations near it so as to prevent vibrations being transmitted from the engine foundation to them.

EXHAUST PIPES.—With the horizontal type that part

of the piping which is above the floor line is usually water jacketed to prevent radiation of heat to the engine room. With the vertical type the manifold and sometimes other connections are also water jacketed. Between the engine valve box or outlet and the silencer the exhaust pipe should have an area of 1.15 to 1.3 times the area of the exhaust valve. From the silencer to the atmosphere with four-cycle engines the size of piping may be slightly decreased but with two-cycle type the area of the exhaust pipe should be as large as possible. Arrangements should be made to allow for the expansion and contraction of the exhaust pipe near the engine and where it becomes greatly heated in operation. In every installation a test cock should be inserted in the piping of each cylinder so that the color, etc., of the exhaust from each cylinder can be tested whenever required.

AIR INLET pipes should be of the following sizes for four-cycle engines:

$$f = \frac{F \times c}{120}$$

For two-cycle engines:

$$f = \frac{F \times c}{60}$$

when f = area of inlet pipes in square inches;
F = area of piston in square inches;
c = piston speed in feet per second.

Cooling Arrangements.—The heat balance of some hot surface engines shows that about 35 per cent of the total B.t.u. supplied in the fuel is carried away by the cooling water, and therefore, the importance of proper arrangements to accomplish this process efficiently is evident. The different methods of cooling are: (*a*) continuous water supply passing through the cooling jackets and then going to waste. (*b*) re-cooling the water (where it is scarce or expensive but of good quality) by means of an open screen type of cooling tower, of simple construction, or other similar apparatus which will accomplish the same result. A considerable amount (one-half to one gallon per B.H.P. hour) of make-up water will have to be added and this should be of such quality as to have no tendency to formation of scale in the passages of the engine. (*c*) use of clean sea water pumped so as to allow of rapid circulation through the passages of the engine. (*d*) enclosed cooling system, an arrangement of which, as recommended by the De La Vergne Machine Company, is shown in Fig. 26. This system is used where only hard, scale-forming or otherwise poor cooling water is available. All the cooling piping and engine jackets are first filled with soft water. The heated water, after discharge through a visible outlet into the sump tank, is pumped through a series of cooling coils and then up to the overhead tank. The coils are showered outside with any available water and only the cooling water of suitable composition circulates within the engine passages. With this system only a slight amount of make-up rain or soft water is required and where necessary

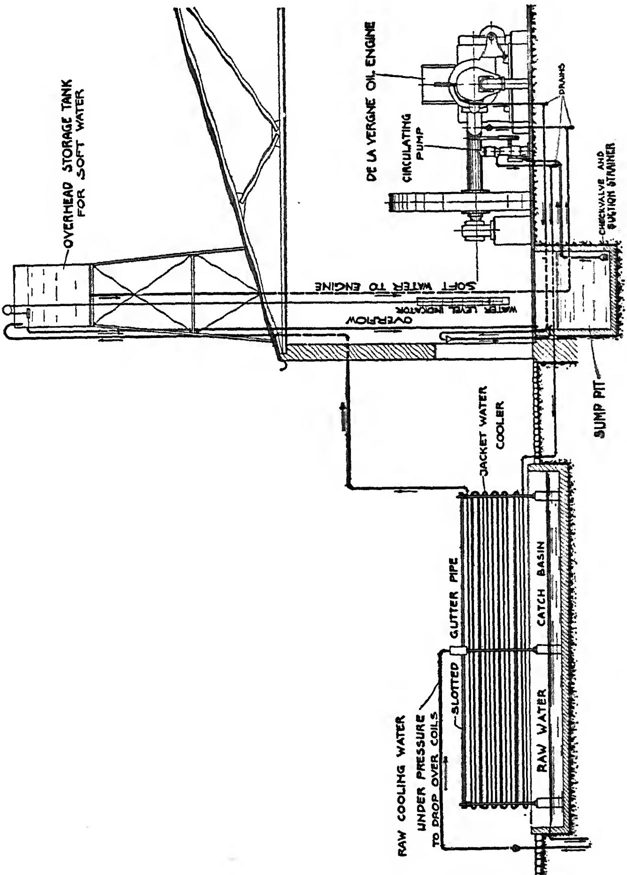

OVERHEAD STORAGE TANK
FOR SOFT WATER
DE LA VERGNE OIL ENGINE
CIRCULATING
PUMP
DRAINS
CHECKVALVE AND
SUCTION STRAINER
SOFT WATER TO ENGINE
WATER LEVEL INDICATOR
OVERFLOW
SUMP PIT
JACKET WATER
COOLER
GUTTER PIPE
CATCH BASIN
SLOTTED
RAW WATER
RAW COOLING WATER
UNDER PRESSURE
TO DROP OVER COILS

FIG. 26.

the outside cooling water can itself be re-cooled in an open cooling tower. Scale formed in the cooling jackets of the cylinder, head or other parts is very injurious; it prevents proper cooling effect and is frequently the cause of cylinder head cracking and other troubles. When scale is found to have been formed it can be removed by filling the passages with a mixture of one-half water and one-half muriatic acid and allowing them to remain filled. After effervescence ceases more acid should be added and if no further action is seen this indicates that the scale has been removed. The amount of cooling water necessary will vary with different designs of engines. The outlet water temperature as recommended by some makers should be about 120° F. and with others may be as high as 180° F. Usually with the initial temperature of the water at 50° F. and a discharge temperature less than 160° F., with a four-cycle engine, 4 to 7 gallons per B.H.P. hour is sufficient. The outlet temperature with a two-cycle engine should not exceed 100°-120°, and then at least 10 gallons per B.H.P. hour should be furnished. All cooling-water devices should be of ample capacity so as to allow of furnishing additional cooling medium in emergencies. With all arrangements the cooling water should flow from a tank, placed 20 feet or more above the engine, to which the cooling water is supplied and from which it gravitates to the various parts of the engine. With any system the flow of water from each outlet should always be visible to the attendant and thermometers should be placed wherever necessary so that proper temperatures are

always maintained in operation. The cooling water should never be cut off immediately after the engine is stopped. Circulation should continue until the parts of the engine have cooled off. Cold water should not be turned on suddenly to an engine already heated. In frosty weather where the engine room is insufficiently heated, water jackets, etc., should be drained to prevent freezing when the engine is standing idle.

SILENCER.—A cast-iron silencer having a cubical area of 6 to 8 times that of the piston displacement is used where complete silencing is unnecessary. When the noise of the exhaust is objectionable larger silencers must be installed.

OPERATION AND CORRECTION

GENERAL REMARKS.—With the modern hot surface type of oil engine, the attendant, to obtain the best results in operation, should first carefully read the instructions accompanying his engine; he should also fully understand the principles of his engine and how to procure proper combustion in the cylinder. The explosive mixture in all designs of these engines consists of (a) hydrocarbon vapor introduced into the vaporizer or combustion space as already described in different ways in the various engines and (b) atmospheric air also introduced into the cylinder by the outward movement of the piston in the four-cycle type or impelled into the cylinder from the scavenging air pump or from the crankcase, in the crankcase compression

type of two-cycle engine, at about 2 to 6 pounds pressure. This mixture of vapor and air, is then suitably compressed by the inward movement of the piston, and is ignited by contact with the hot surface (hot tube, electric coil or otherwise), or by compression only, and the impulse or power stroke of the piston follows. The exhaust period afterward takes place. If these conditions do not properly exist correct operation will not follow and the cause for improper working may be found by examining the following points. (This subject is also discussed in Chapters VII, VIII and IX of Part II.)

IGNITION in all types is dependent on: (a) proper pressure of compression in the high pressure type or on this condition together with proper temperature of the hot surface or other device in the low pressure engines. Insufficient pressure is probably due to leakage of some moving part or valve. Insufficient temperature of igniting part may be traced to (a) too great a cooling water circulation, (b) to carbonized or unclean surfaces, or (c) to possible water leakage into combustion space.

OIL SUPPLY to the cylinder should be correctly timed (see diagrams Figs. 22 and 23 and also the instructions for operation of the particular make of engine). The correct quantity of fuel must be furnished; (a) too much oil makes the mixture too rich and will prevent proper combustion, and (b) too little oil will make the mixture too poor. The color of the exhaust gases will frequently indicate whether the oil supply is correct. Too much oil will usually entail black or easily visible

exhaust. The exhaust gases should be invisible or nearly so. Too small a supply of fuel will be shown by inadequate power being developed.

THE AIR SUPPLY in the four-cycle type is dependent on proper opening and closing of air inlet valve and on absence of leakage by the valves or piston; in the two-cycle type it is dependent on absence of leakage in the crankcase or air pump, and in either type obstruction of air passages may prevent proper air supply. A low-pressure indicator card should show the cause if the air supply is incorrect.

OIL SPRAY to the vaporizer, etc., should be sharp without drip and should be such that the vapor or globules of petroleum are thoroughly distributed throughout the whole space of the vaporizer or combustion space. The vapor should be intimately mingled with air and globules should have their maximum surface exposed to the oxygen. The speed of the spray as it enters the combustion space and the correct period during which it enters this space are important and are generally controlled by the shape of the cam actuating the fuel pump and by the size of the spray nozzle. Diagrams Figs. 22-23 show the proper period of fuel injection in representative types of engines.

KNOCKING in the hot surface oil engine may be caused by (a) loose bearings in the connecting rod or in the main crankshaft bearings, (b) loose flywheel keys, (c) improper timing or ignition which may be due to too high compression pressure or to too early injection of the fuel or to operating with insufficient cooling water circulation. Indicator cards taken with

full load should show the cause in any event. The ignition line should appear perpendicular to the atmospheric line as seen in Fig. 58.

Deficient power may be due to a variety of causes: (*a*) increased friction in the moving parts caused by improper lubrication; (*b*) sticky or gummy deposit on the piston, which, when operating properly, should be clean, with proper film of clean lubricant surrounding it; (carbonized or clogged piston rings will allow carbon to be blown past them and may then require removal or cleaning); (*c*) improper cooling effect which allows the piston to expand unduly, causing greater friction, and possibly to become distorted; (*d*) overheating of other moving parts, bearings, etc., causing greater friction and loss of power; (*e*) inadequate fuel supply due to leaky fuel inlet valves or leaking pump valves or plungers; (*f*) a clogged oil filter or a pocket of air in the fuel supply connections causing improper supply of fuel; (*g*) improperly timed ignition of the explosive mixture; (*h*) a leakage of the air inlet or exhaust valves or piston rings causing loss of compression pressure. Careful examination of the parts above referred to should enable the attendant to ascertain the reason for deficient power, but indicator cards must be taken if the cause cannot otherwise be located.

Piston blowing may be due to various causes, (*a*) improper lubrication or use of lubricant unsuitable for this special purpose, as specified elsewhere; (*b*) piston rings which have become carbonized and stick in their grooves, or have become worn so that they do not make a tight fit in the cylinder liner; (*c*) piston and rings

scored or cut by grit or dirt which has come in contact with them. (*d*) overheating and uneven expansion caused by insufficient water cooling. If the blowing of the piston cannot otherwise be remedied new piston rings must be put in place and in some cases it is even necessary to rebore the cylinder and supply a new piston and rings. Occasionally lubricant splashing against the heated back end of piston and vaporizing will cause vapor to issue from the open piston end. This may be taken erroneously to be leakage past the rings.

Leakage of air inlet and exhaust valves will of course result in defective operation. In the smaller sizes an engine can be turned backwards by hand on the compression stroke and if these valves and the piston are tight it will be impossible or difficult to turn the piston past the inner dead center. In larger engines tightness of the exhaust valve can be tested by turning the crank till the piston is on the inner dead center at the beginning of the combustion stroke. The valve can then be opened and the air from the starting tank allowed to fill the combustion space and valve chambers. The engine cannot start, being exactly on dead center, and any leakage past the exhaust valve will be heard through the test cock opened on the exhaust piping. A leaky valve should be taken out and ground in with carborundum power till the seat is perfect all around it. If necessary and if the exhaust valve is badly pitted it must be trued up in a lathe.

Lubricants.—The main cylinder and piston should be lubricated with a light mineral oil having a viscosity

of 300° to 750° Saybolt at 100° F., having a high flash point (about 370° F.) and free from animal matter. The air compressor cylinder should have little or no lubrication. When any lubricant is applied it should be similar to that used on the main piston.

CHAPTER IV

TWO-CYCLE OIL ENGINE—DESCRIPTION OF VARIOUS DESIGNS

THE TWO-CYCLE type sometimes referred to as the "two stroke" or "two-stroke cycle" oil engine is at present manufactured largely in England, Sweden, and other countries of Europe as well as in the U. S. A. It is built in sizes up to 125 B.H.P. in one cylinder having about 20″ diam. and 30″ stroke, but this type is more generally used in the smaller sizes from about 6 to 50 B.H.P. in one cylinder. With the multi-cylinder designs three and four cylinders are employed, both for stationary and marine purposes. Oil engines of smaller powers are in service more in Europe than in the U. S. A., because in the former countries there is greater demand for small isolated power plants and gasoline (or petrol) is more expensive, while in the latter country the small size gasoline engine is most extensively used. Electric current from hydro-electric and other economical central generating stations is also available more generally and the electric motor is frequently installed. For small marine purposes, the gasoline engine is also in general use in the United States. The chief feature of the two-cycle engine is

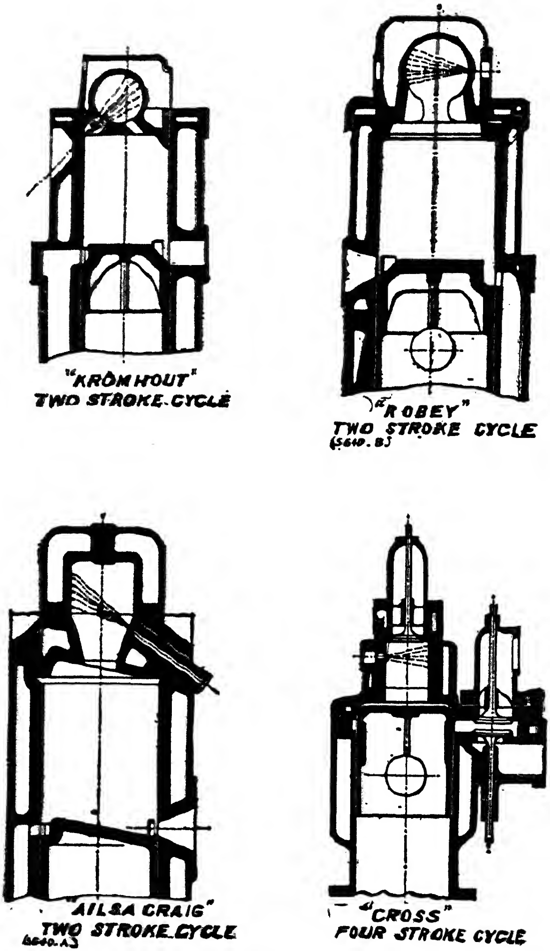

"KROMHOUT"
TWO STROKE CYCLE
"ROBEY"
TWO STROKE CYCLE
(5640. B)
"AILSA CRAIG"
TWO STROKE CYCLE
(5640. A)
"CROSS"
FOUR STROKE CYCLE

Fig. 27.

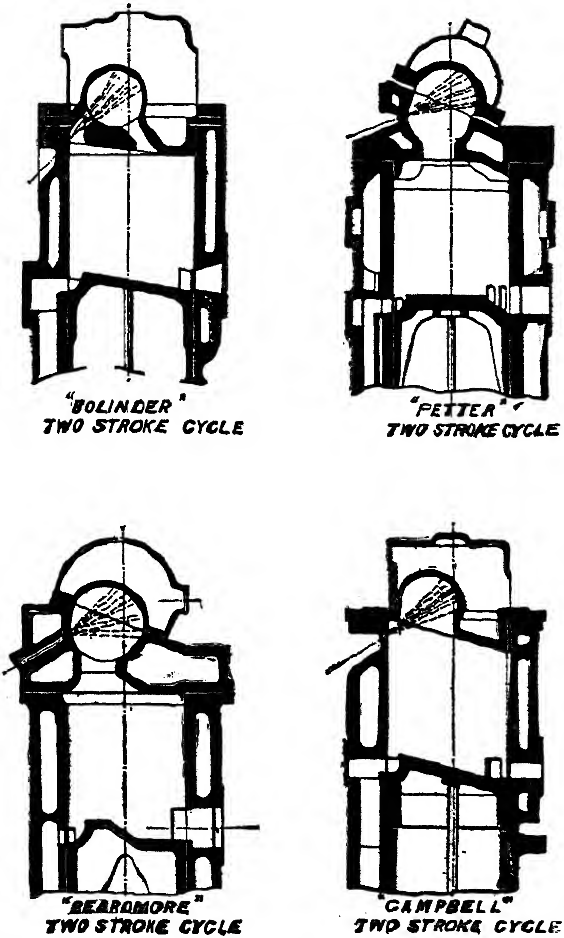
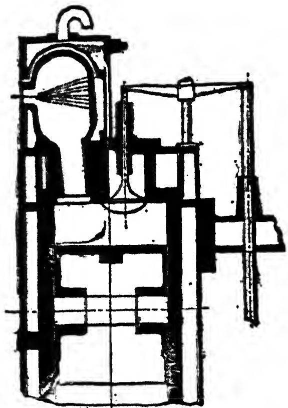


FIG. 27—*continued*

its simplicity of construction and the absence of air and exhaust valves necessary with the four-cycle engine. The weight per B.H.P. is only about 70 per cent of the four-cycle type. The two-cycle type is also referred to on page 17 of Part II and its method of operation is there outlined.



"HEIN"
FOUR STROKE CYCLE

FIG. 27—*continued*

VARIOUS TYPES.—At Fig. 27 are shown the arrangement of cylinders and "hot bulbs" or hot surfaces of the leading European two-cycle oil engines.* Each diagram has the name of the builder indicated beneath it.

*Reproduced from Mr. James Richardson's paper read before the Diesel Engine Users' Association, London, 1918. Figs. 27 to 31 are also reproduced from the same source.

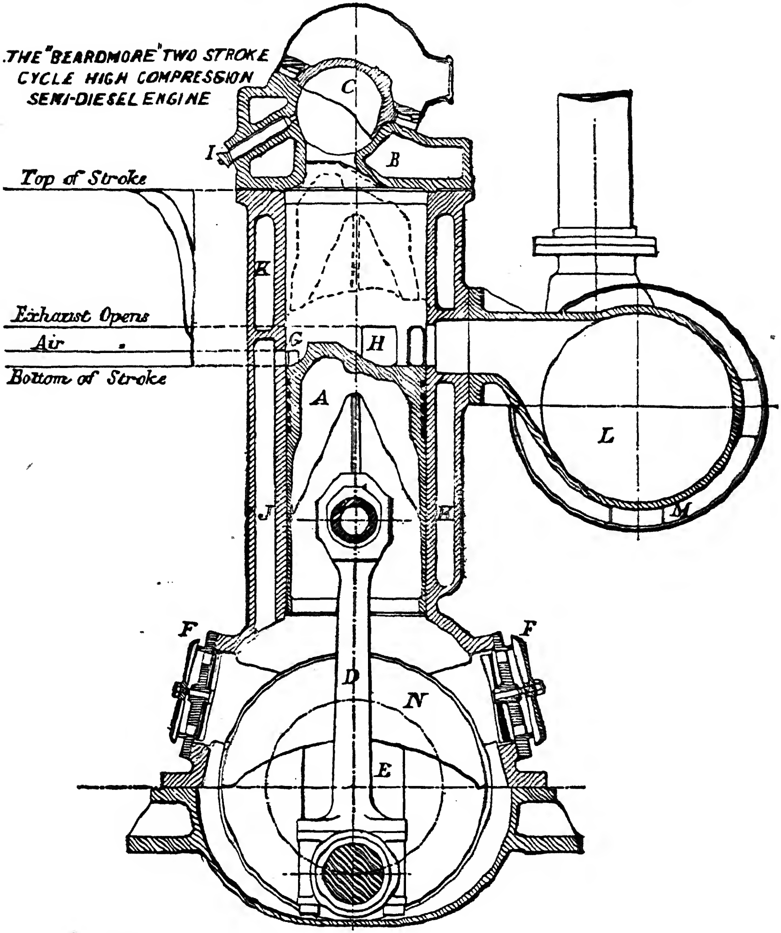


FIG. 28.—A. Piston. B. Cylinder cover. C. Combustion chamber. D. Connecting rod. E. Crank. F. Air inlet valves. G. Scavenging air ports. H. Exhaust ports. I. Fuel injection nozzle. J. Scavenging air passage. K. Water jacket. L. Exhaust silencer. M. Water jacket. N. Balance weight.

In Fig. 28 is shown the Beardmore two-cycle oil engine. The parts of the engine are also referred to below and an indicator diagram at the side of the cylinder is reproduced on a small scale.

PRESSURES.—Fig. 29 shows a typical indicator diagram from a two-cycle oil engine with a crankcase compression. The pressure of compression in the cylinder is approximately 185 lbs., and the maximum pressure about 300 lbs. Taking into consideration

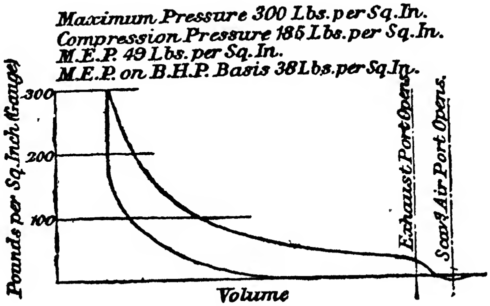


FIG. 29.

all features of design, including theoretical and mechanical efficiency, bearing pressures, cyclic regularity and first cost, compression pressures of 150 lbs. to 185 lbs. seem by experience to be the most advantageous pressures which are employed in this type. The flexibility of the two-cycle oil engine, a very important requirement especially with marine installations, may be considered under the following headings:

(*a*) Constant M.E.P. with varying R.P.M. and consequent varying power developed.

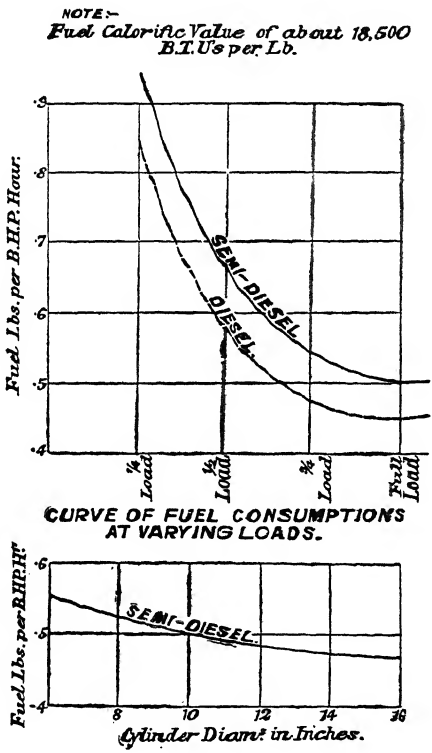

NOTE:—
Fuel Calorific Value of about 18,500 B.T.U's per Lb.
Fuel Lbs. per B.H.P. Hour.
·9
·8
·7
·6
·5
·4
SEMI-DIESEL
DIESEL.
¼ Load
½ Load
¾ Load
Full Load
CURVE OF FUEL CONSUMPTIONS AT VARYING LOADS.
Fuel Lbs. per B.H.P.H.
·6
·5
·4
SEMI-DIESEL.
8
10
12
14
16
Cylinder Diamr. in Inches.

FIG. 30.

(*b*) Constant R.P.M. with varying M.E.P. and varying power developed.

(*c*) Varying R.P.M. and varying M.E.P.

Condition (*a*) is considered impractical as a reduction in R.P.M. adversely affects scavenging and compression and accordingly lowers the M.E.P.; in condition (*b*) the volume of air entering the crankcase remains practically constant and the scavenging effect is, therefore, nearly constant but at a lower temperature and pressure. Condition (*c*) is accomplished without difficulty.†

THE FUEL CONSUMPTION of this type of engine is shown by the curves in Fig. 30 and in comparison with that of the Diesel engine and that at varying loads is also shown.

THE PERIOD of injection of the fuel in the two-cycle oil engine is shown in the diagram reproduced at Fig. 31 and which also shows the method of allowing a relatively earlier injection period for half load and lighter loads whereby the hot surface or bulb is maintained at the necessary temperature to cause ignition at very light load. In the description of the Petter type this feature peculiar to that type is referred to more fully.

GENERAL REMARKS.—The temperature of the hot surface: with a "hot bulb" is important, if overheated, the high temperature may cause "cracking" of the particles of fuel and tend to split up the heavier hydrocarbons into the lighter hydro-carbons, which results

†Extracts from "Semi-Diesel Oil Engine," Diesel Engine Users' Association, London, 1918.

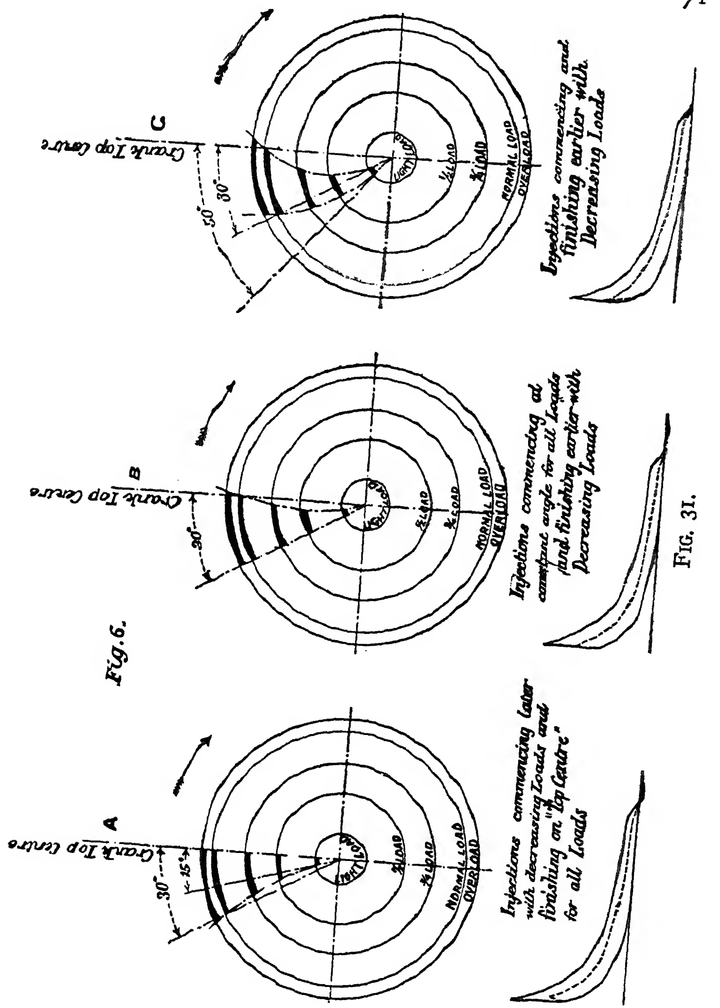

A
B
C
Crank Top Centre
30°
15°
50°
LIGHT LOAD
½ LOAD
¾ LOAD
NORMAL LOAD
OVERLOAD
Injections commencing later with decreasing Loads and finishing on "Top Centre" for all Loads
Injections commencing at constant angle for all Loads and finishing earlier with Decreasing Loads
Injections commencing and finishing earlier with Decreasing Loads

Fig. 6.

FIG. 31.

in the formation of carbon deposits most injurious to satisfactory operation. With the bulb or hot surface at too low a temperature the particles of fuel are only partially gasified or vaporized and the heavier hydro-carbons will be deposited in the form of a gummy substance which will result in "gummed up" piston and rings and which will be found difficult to remove when cold. It is, therefore, very important that the proper temperature of the bulb or surface be maintained, usually this should be from 750° to 1100° F., that is, just "visible red" heat to "dull or cherry red." With tar oils as fuels these temperatures must be higher as those fuels require higher temperatures (and higher compression) than petroleum fuel oils. The M.E.P. in the crankcase-compression two-cycle oil engine is about 48 to 55 lbs. and is most largely governed by the low volumetric efficiency of compression in the crank-case. In most designs of this description the volume of air entering the combustion space does not permit of greater fuel injection than will develop the above pressures. Where scavenging air is compressed in a separate pump a greater volume of air can be injected into the combustion space and consequently higher M.E.P. can be developed.

Piston speeds on these types are recommended at from 700 to 900 ft. per min. With higher speeds the weight of air drawn into the combustion space is insufficient and no advantage is gained by this procedure.

Exhaust.—It is important that the velocity of the exhaust gases through the ports should be high in order to assist the scavenging process and thus ensure

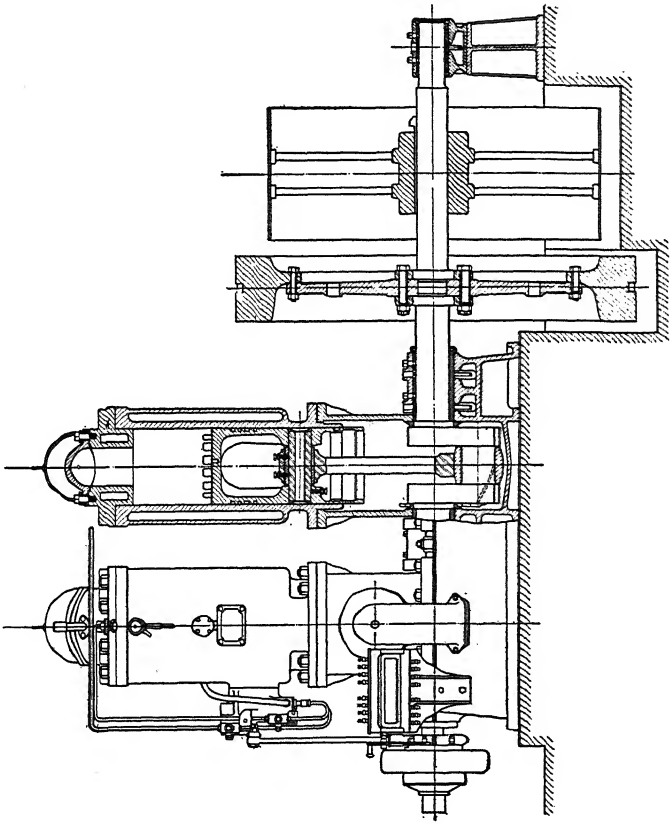


Fig. 32.

the entrance of the maximum amount of air into the combustion space for the following cycle.

THE PETTER vertical two-cycle oil engine with crank case compression is manufactured in large numbers by the Vickers-Petter Co., at Ipswich and at Yeovil, England. In the smaller sizes (Petter junior) the fuel, kerosene, is admitted to the cylinder by aspiration. Electric ignitor with high tension magneto is used, the engine being started with gasoline (petrol). In the larger Petter designs which are all of the two-cycle type, hot surface ignition is employed entirely; the shape of the vaporizing chamber is seen in Fig. 32 and Fig. 35. An external lamp for heating the vaporizer is required for a few moments before starting. Compression of the air to about 6 lbs. in the crankcase takes place during each downward stroke of the piston, air inlet to the cylinder and combustion space being through the ports uncovered by the piston shown in Fig. 35. The fuel pump is actuated from an eccentric operated from the crankshaft direct. The amount of fuel delivered to the sprayer is varied by lengthening or shortening the stroke of the pump, which is effected through a system of floating eccentrics controlled by the governor shown in Figs. 33 and 34. The fuel consumption at various loads is indicated in Fig. 36.

The distinctive feature of the Petter oil engine is the system of variable fuel injection period at lighter loads the fuel injection period commences approx.: 160° before the crankpin reaches the upper dead center. With this arrangement full load or overload can be carried without pre-ignition and then with light or no

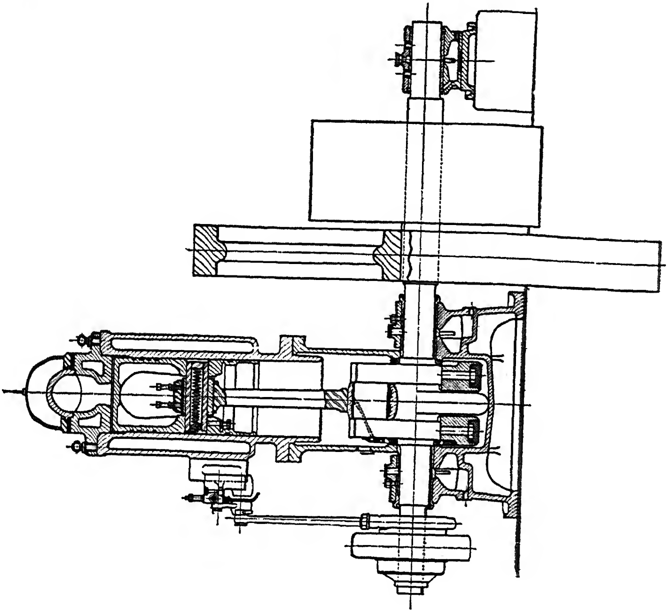


FIG. 34.

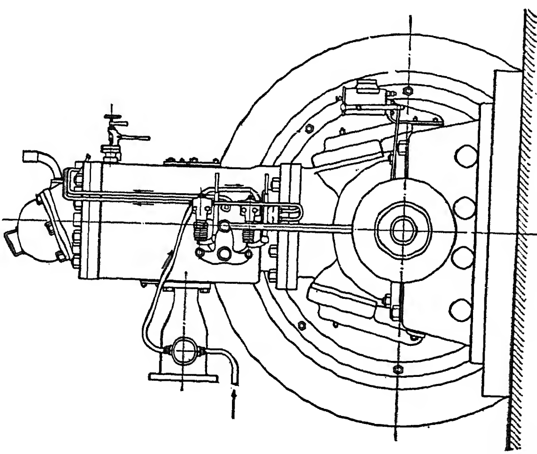


FIG. 33.

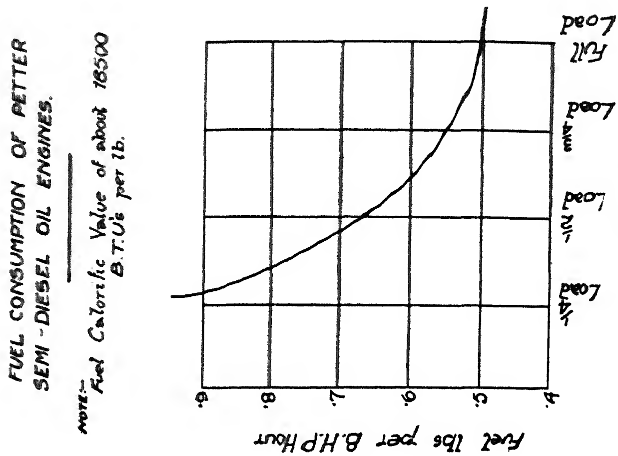

FUEL CONSUMPTION OF PETTER
SEMI-DIESEL OIL ENGINES.
NOTE:-
Fuel Calorific Value of about 18500
B.T.U's per lb.
.9
.8
.7
.6
.5
.4
Fuel lbs per B.H.P Hour
1/4 Load
1/2 Load
3/4 Load
Full Load

FIG. 36.

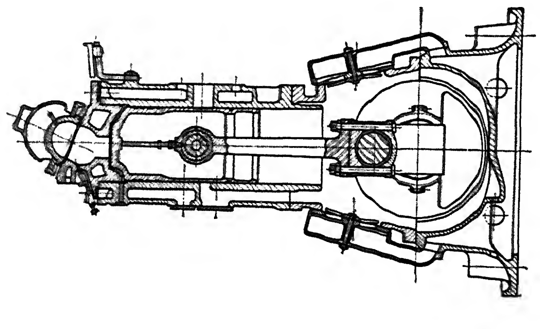


FIG. 35.

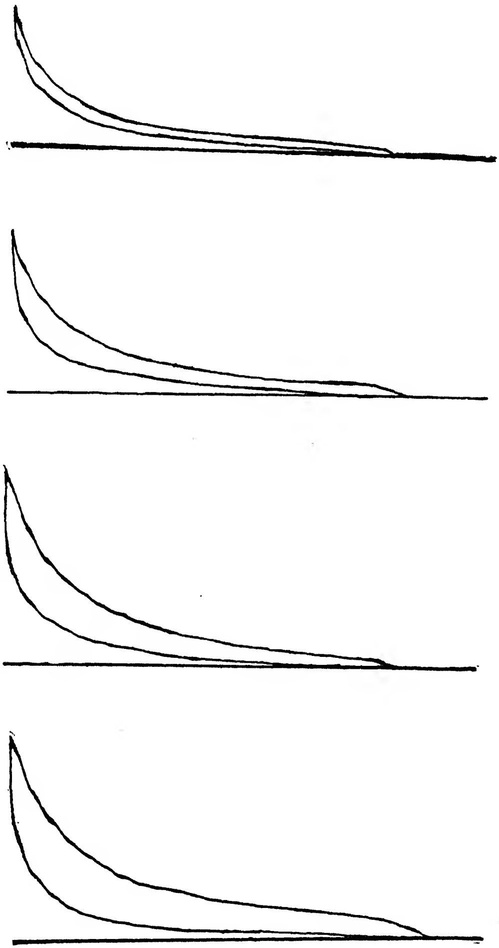


Fig. 37.

load, adjustment can be made so that the earlier period of fuel injection allows sufficient heat to be maintained in the vaporizing chamber.

The compression pressure is approximately 150 lbs. At Fig. 37 are shown indicator diagrams with full load, three-quarter, half and light load. In Fig. 38 are

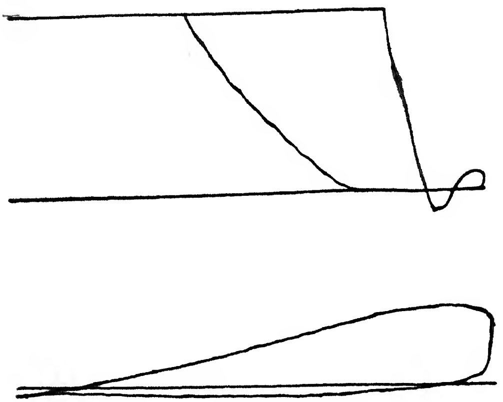


FIG. 38.

shown light spring diagram and crankcase indicator card. The following dimensions of an engine of this design are:

Cylinder bore 12″, stroke 14″, rated H.P. 70, crankshaft diameter 5¼″, flywheel diameter 66″, approximate weight 15,000 lbs. The results of a test of an engine of this type are:

TABLE VI.—TEST ON PETTER OIL ENGINE

	Full load	Half Load	Quarter Load
Brake horse power	71	36.8	18
Revolutions per minute	227	229	231
Total fuel used, pints (Imp.)	62	18	14
Hours run	2	1	1
Fuel per hour, pints (Imp.)	31	18	14
Fuel per B.H.P. Hour, pints (Imp.)	0.437	0.49	0.778
Fuel used	Texas Crude	Texas Crude	Texas Crude
Specific gravity of fuel	0.9	0.9	0.9
Beaumé	26	26	26

MARINE ENGINE.—The Petter oil engine is also largely used for marine purposes. With the small sizes a geared reversing motion is used, but with the larger units a reversing motion is employed by which the direction of rotation of the crankshaft is reversed in the following sequence of movements:

(1) The air valve is opened, (2) the fuel supply pump is put out of action by means of the hand lever, (3) the engine slows down then the reversing lever is moved to "action," thus admitting air pressure above the piston which then descends operating the crankshaft in the reversed direction of rotation, (4) the fuel pump is put in motion again and the engine operates reversed.

THE FAIRBANKS MORSE two-cycle vertical three-cylinder marine oil engine, largely manufactured in the United States is shown in section at Fig. 39. It operates on comparatively low compression pressure, the ignition being caused by hot surface. The solid construction of the frame A and the main crankshaft bearings carried by it are shown in the illustration. The cylinder B and cylinder head C are both maintained at an even temperature by liberal water jacketting. The hot surface or "bulb" into which the fuel is injected and which causes the ignition is secured directly to the cylinder head being held in place by the vaporizer ring D. At starting this part is heated by external heating lamp for a few moments in the usual way.

The fuel pump E is operated directly from the crankshaft by cams as shown. Each cylinder is fed by separate fuel pump. The crankshaft with its crankpins

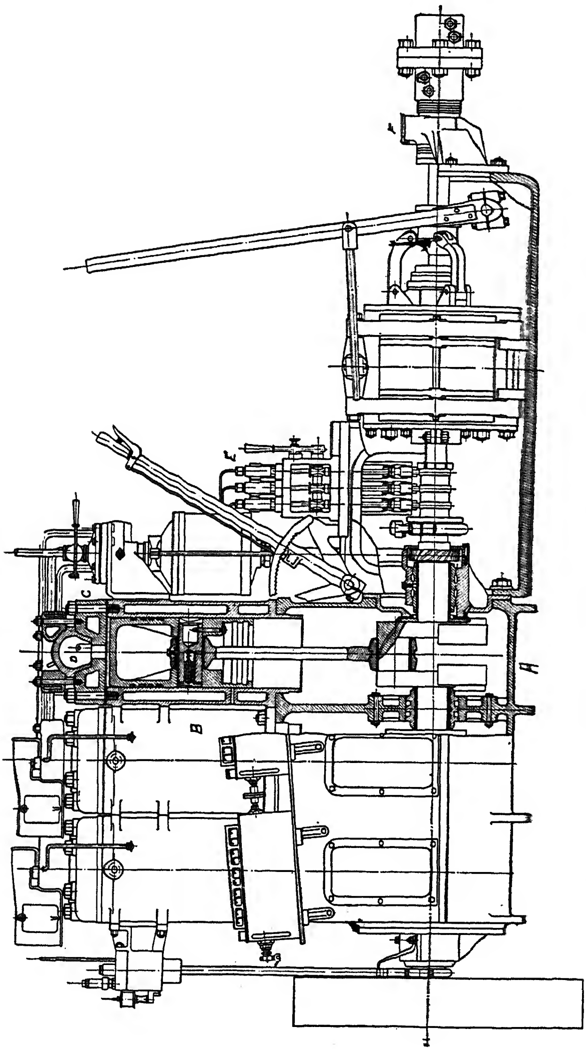


Fig. 39.—B, Cylinders, C. Cylinder head. D. Vaporizer. E. Oil supply pump.

at 120° is carefully counterbalanced. The thrust bearing F is carried on an extension of the crankcase. Thorough lubrication to all parts requiring it, is effected by means of force feed lubricator mechanically operated. The fuel inlet or spray nozzle which is very efficient in action is shown in section at Fig. 10. The fuel first enters through the pipe connection C screwed into the socket B which in turn is secured by thread into the part A which is inserted directly into the wall of the vaporizer. The fuel first passes the ball check valve F held on its seat by the spiral spring G. The fuel then passes to the filter D held in place by contact of socket at E. The liquid fuel afterwards passes through the spiral passages cut in the periphery of the second valve stem H held on its seat by the spiral spring K. The lift of this valve is controlled by the space allowed before it comes in contact with the tip J screwed into the part A. In this tip J is drilled the nozzle I through which the fuel is forced in minute spray form.

THE BESSEMER horizontal two-cycle engine is built with single cylinders from 15 to 85 H.P. and is illustrated at Fig. 40. The air inlet is taken in through the main engine frame casting, and passes through the Corliss type valve, shown in the sectional view which is actuated from an eccentric placed on the crankshaft. Cooling water is arranged to pass under the crankcase and the lower crosshead guide and thence to the cylinder water jacket. This arrangement cools the lubricant in the crankcase which lubricates the main bearings, connecting rod, cross-head and wrist pin bearings.

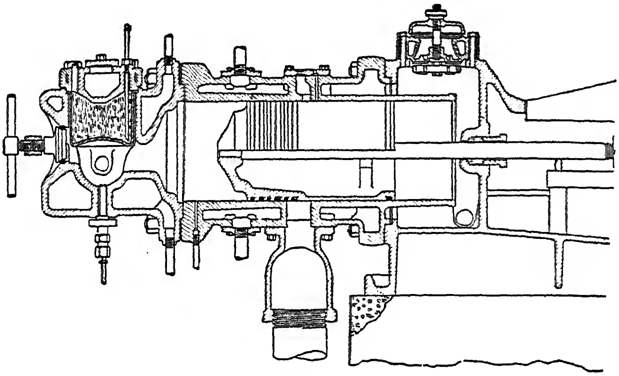


FIG. 40.

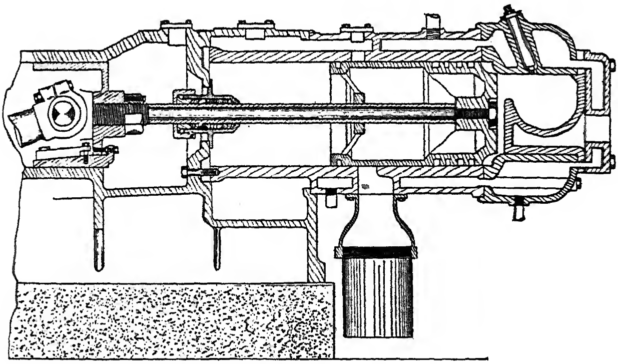


FIG. 41.

The vaporizer casting in the cylinder head gives the necessary clearance for the low compression used and permits the location of the spray valve in such a position as to allow the injection of the fuel oil directly up against the uncooled part of the vaporizer. The vaporizer is heated before starting by a blast lamp. The governor is arranged to give a variable fuel pump stroke, and the engine speed is further regulated by an auxiliary cam-actuated linkage connected to the governor which holds the pump suction valve open except for a variable period of 12° to 16° of the crank angle, during which the valve closes suddenly and fuel is injected into the combustion chamber.

In the smaller size engines the combustion chamber is made of steel forging and contains approximately three pounds of mercury which is completely sealed in the interior of the vaporizer. The object of the mercury is to maintain constant vaporizer temperature irrespective of the load on the engine.

The fuel economy of the Bessemer engine varies according to the cylinder size: At full load from 0.65 to 0.9 pound, at three-quarters load from 0.7 to 1.0 pound, at half load from 0.8 to 1.2 pounds. Cylinder dimensions range from 8½" and 15" stroke to 16" diameter and 20" stroke. The speed of this engine varies from 275 to 250 R.P.M. The weight of the engine per brake horsepower for single cylinder units ranges from 390 to 275 pounds.

THE BUCKEYE two-cycle low-compression horizontal oil engine is illustrated in Fig. 41. Air for scavenging enters through openings in the front end of the crank-

case and passes under the frame through ports and intake plate valves on the engine side near the crosshead, then into the combustion space behind the piston. A box-shaped volume in the lower part of the frame and connected to the above space is proportioned to give a scavenging air pressure of about 3 pounds per square inch. The trunk piston has a length sufficient to close the air inlet and exhaust ports on the inner dead center. The piston rod connecting with the crosshead, passes through a stuffing box which seals the scavenging air space in front of the piston. The crosshead and crank pin bearings work in the enclosed crankcase permitting splash lubrication for these parts. The cylinder jacket and liner constitute an integral casting, the two being connected by walls forming the intake and exhaust ports and by a flange at the head end through which are the openings for the passage of cooling water into the cylinder head. Cooling water is admitted to the jacket at a point near the exhaust port. Only that section of the cylinder extending from the above ports towards the head end is water-jacketed. The cylinder extends back to a flange which is secured to the main frame by studs.

A simple check-valve-type spray valve injects oil into the center of the cup-shaped uncooled vaporizer. Absence of other valves in the head design permits of this construction. For starting, the latter is heated by a lamp.

When using fuel or crude oil of about 30° Bé. (having lower heat content of approximately 18,500

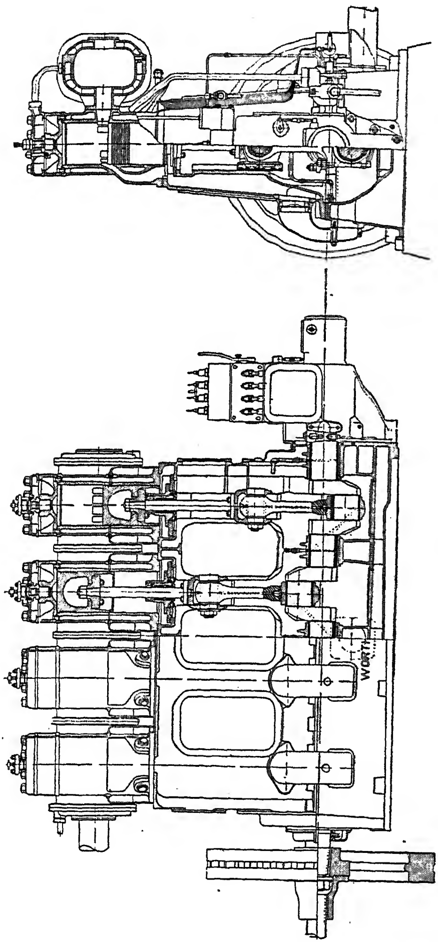
B.t.u.), the fuel consumption is as follows under ordinary operating conditions:

Full load:	0.65—0.7	pound per B.H.P. hour;
Three-quarter load:	0.70—0.75	" " " "
Half load:	0.80—0.90	" " " "

The total weight per B.H.P. varies from 270 to 320 pounds, being governed by the size of engine, the arrangement of cylinders, and speed. Lubrication of parts other than the crank pin, wrist pin and crosshead bearing surface is from a mechanical forced feed lubricator. Engine speed is controlled by a simple governor which acts on a by-pass valve placed in the discharge side of the fuel supply pump.

THE WORTHINGTON two-cycle, solid or airless injection, vertical Diesel oil engine is shown in section at Fig. 42. This is a recently developed type and is built in sizes from 30 to 300 B.H.P., the former in single cylinder, the latter in four-cylinder units for stationary installations. For marine purposes four-cylinder designs from 75 to 300 B.H.P. are also built. The construction illustrated in Fig. 42 shows the piston made of sufficient length to close and seal the ports shown in the cylinder walls, the piston rod passes through a stuffing box and crosshead and guides are provided. The crank end of the main cylinder is employed as a scavenging pump furnishing air at approximately 6 lbs. per sq. in. pressure to the main combustion space. The scavenging air compression chamber is outside and entirely separated from the crankcase. This arrangement permits of the use of force feed lubrication and

TWO CYCLE—SOLID INJECTION



LONGITUDINAL AND TRANSVERSE SECTIONS

Fig. 42.

avoids the loss of lubricant through its being swept along with the scavenging air.

In a test on an experimental engine having cylinder 10¼" diameter and 11½" stroke operating at 375 R.P.M., a fuel consumption of 0.439 lbs. at full load and 0.536 lbs per B.H.P. hour at half load was recorded.* Fuel injection (without air blast) takes place just before the piston reaches the inner dead center. The fuel spray first enters the "injection chamber" which is shown placed above the main cylinder and combustion space in Fig. 42. The fuel injection period lasts about 15° of the crank angle. The fuel is supplied to the sprayer by the fuel pump actuated by eccentric attached to the crankshaft. The four-cylinder 300 B.H.P. engine (Fig. 42) has cylinders each 15¼" diameter and 16" stroke with 240 revolutions per minute. Indicator diagram from the main cylinder is reproduced at Fig. 43 and that from the scavenging chamber is shown at Fig. 44.

*"Motorship," November, 1921.

2 CYCLE SOLID INJECTION.
TYPICAL INDICATOR CARDS
AT FULL LOAD

Working Card
375 RPM
Spring: 1" = 500 lb/sq.in.

FIG. 43.

Scavenging Card.
Spring: 1" = 15 lb./sq.in.
Taken at the same load as the working card above.

FIG. 44.

CHAPTER V

FOUR-CYCLE OIL ENGINES—DESCRIPTION OF VARIOUS DESIGNS

PRELIMINARY.—The two-cycle engines discussed in the previous chapter are used more for work where varying load, intermittent service or where occasional stoppage of the engine can be allowed, while for continuous service day and night for long periods at or near full load, or where low grade fuels are used the four-cycle, slow speed oil engine of heavy construction is preferred, notwithstanding the fact that for such service the two-cycle engine is sometimes installed. Engines to give satisfactory service under such conditions should have comparatively low bearing pressures and ample and efficient lubricating devices so that in the emergencies arising in continuous operation, heated bearings or other surfaces can be flooded with lubricant temporarily and shut-down prevented. In such engines, again, an overload capacity is usually (or should be) provided available without the necessity of over-heating the combustion space (by the injection of too great amount of fuel) so that power can be developed when necessary, to overcome the greater friction in the temporary emergencies above referred to. An ideal installation allows at least 10 per cent of the

engine's rated capacity to be held in reserve and always available for any emergencies. The two-cycle type is manufactured chiefly of the vertical design while the four-cycle type has hitherto been built more largely of the horizontal design. Some builders are now favoring

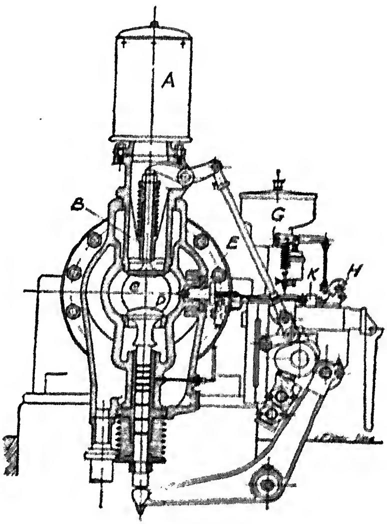


FIG. 45.

the vertical type and a difference of opinion exists as to their relative advantages. The horizontal design allows greater accessibility to valves, valve motion, piston and connecting rod bearings. The vertical type occupies about two-thirds of the floor space of the horizontal type and it requires somewhat less concrete

in the foundations as the free force is in the vertical plane. With multi-cylinder vertical engines the cost of production should be less than with the single cylinder horizontal design.

THE CROSSLEY Oil Engines for use with either refined or crude and residual petroleum oils, tar oils, etc., are made in sizes from 20 to 130 B.H.P. per cylinder, all of the four-cycle horizontal type. The following description applies to all sizes and the illustrations are made from drawings of the engine having cylinder bore of 18.5 inches and stroke 28 inches which developed 117 B.H.P. at 180 R.P.M.

Fig. 45 shows the engine in section through the combustion chamber. Fig. 46 shows the engine in section through the cylinder. Fig. 47 illustrates the oil pump and oil control details, while Fig. 48 shows the construction of the oil heater as used for heavy oils. Figs. 49, 50 and 51 illustrate certain physical actions in the combustion chamber.

On the suction stroke air is drawn in through the air silencer A, air valve B and combustion chamber C into the cylinder. Towards the end of the compression stroke a charge of atomized fuel is injected by pump J through the oil sprayer E into the compressed and heated air in the combustion chamber where it is ignited. The burnt products are driven out of the cylinder through valve D on exhaust stroke of the piston.

Three physical rather than mechanical operations take place in rapid succession during the cycle.

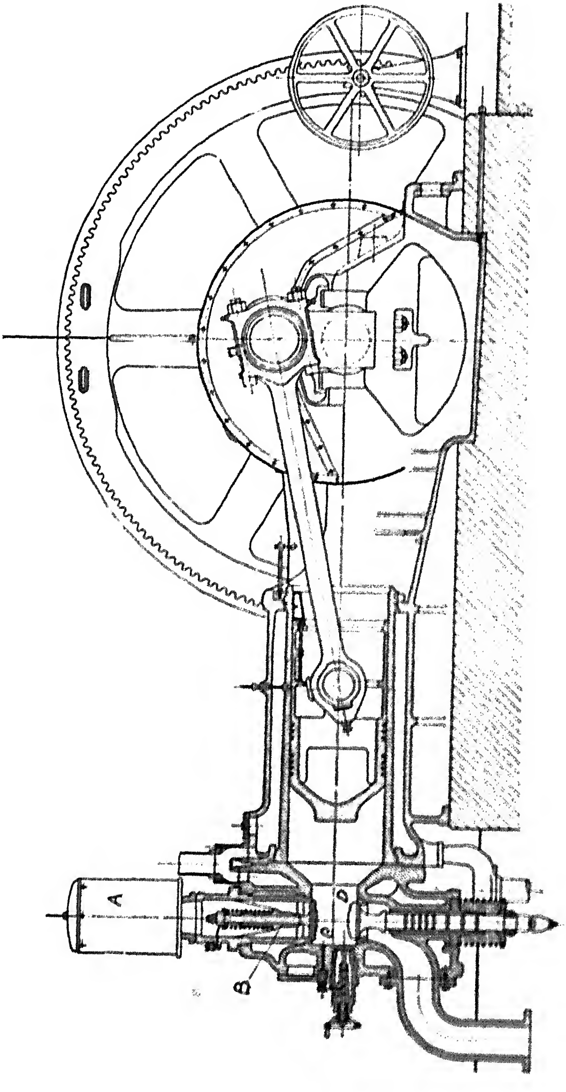


Fig. 46.

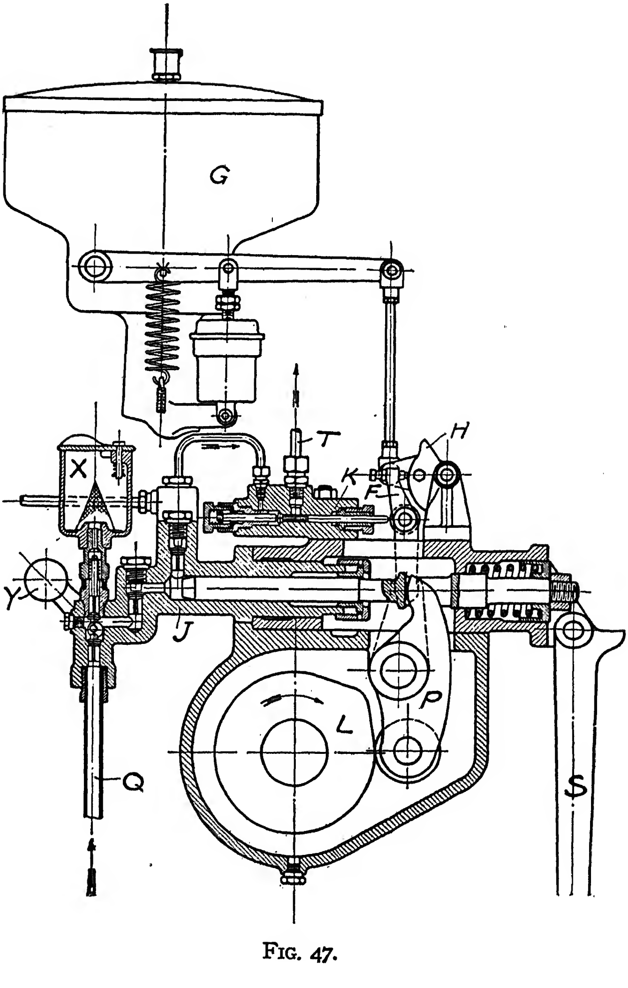


FIG. 47.

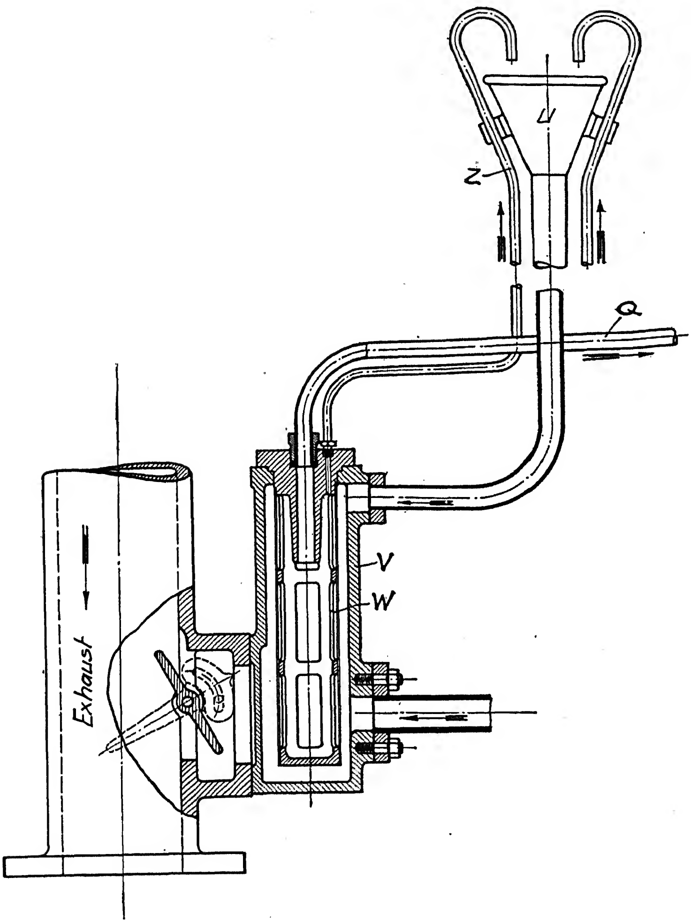


FIG. 48.

(1) Complete scavenging of the burnt products. At the end of the exhaust stroke both air inlet and exhaust valves are then open and the inertia effects of the column of exhaust gases in the exhaust pipe thus increases the volume of air in the combustion chamber and by its cooling action also increases the weight of the air contained thus enabling the engine to work cooler, and a greater overload to be obtained. This scavenging action is illustrated in Fig. 49. (2) When the piston is at the end of the compression stroke the combustion chamber allows the bulk of air to be in a very compact mass and elongated in the direction in which the oil is sprayed. This distance, as shown in Fig. 50, is greater than the distance across the combustion chamber. Such condition counteracts the tendency to cool the central zone of air mass where the temperature is greatest. By making the chamber elongated the hottest zone is automatically moved from the center of the chamber to a point somewhat further from the sprayer as indicated by a dotted circle in Figs. 50 and 51. It is claimed that this enables the compression pressure to be somewhat reduced without impairing the efficiency of the engine starting cold. (3) When the spraying of the oil into the combustion chamber is in full blast the projection shown on the end of the piston enters the combustion chamber (see Fig. 51), and the imprisoned air in the cylinder is then forced through the space between it and the chamber walls creating considerable turbulence, thus improving the processes of vaporizing the oil, mixing the fuel with the air, and burning the mixture. Turbulence

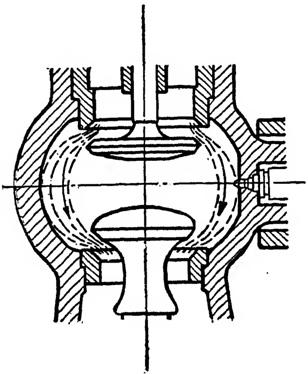


FIG. 49.

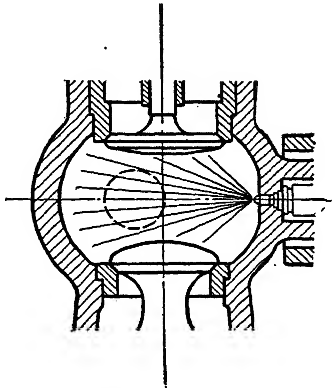


FIG. 50.

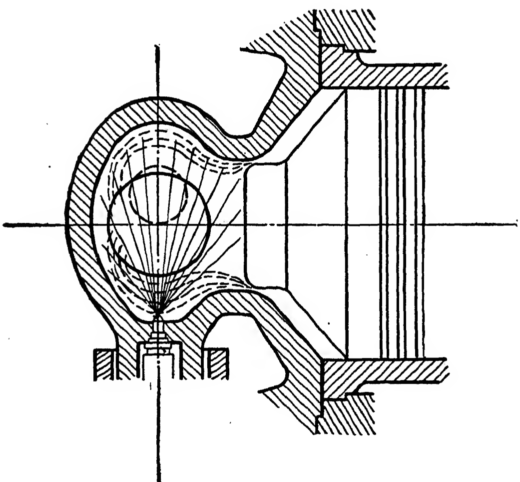


FIG. 51.

takes place mainly nearest the walls of the chamber so that the hot "zone" is not affected until after ignition has taken place.

The oil pump J is operated by means of a cam L through a lever P. The cam runs in a lubricating oil bath.

The governing of the engine is effected by the opening of a by-pass or control valve K in the oil fuel pipe between the oil pump J and the oil sprayer E at variable points in the pump stroke according to the load. For instance, at medium load the valve is opened at about half pump stroke, the remainder of the oil in the pump, instead of passing into the combustion chamber, is by-passed through the control valve K and pipe T shown in Fig. 47. The floating lever F opposite the end of the control valve spindle is pivotted at its bottom end to the pump lever P. The upper end of the floating lever rests against a quadrant H, the position of which is controlled by the governor G. The quadrant has a cam-shaped periphery and as the governor rises and falls the upper end of the floating lever is pushed to and fro in a direction towards or from the end of the control valve spindle.

When using heavy fuels which do not readily atomize at normal temperatures a change-over fuel device is provided to enable a lighter fuel to be used for a few seconds or minutes when starting the engine. This consists of an eccentric pin in the fuel pump placed between the two suction valves and operated by the weighted hand lever shown at Y in Fig. 47.

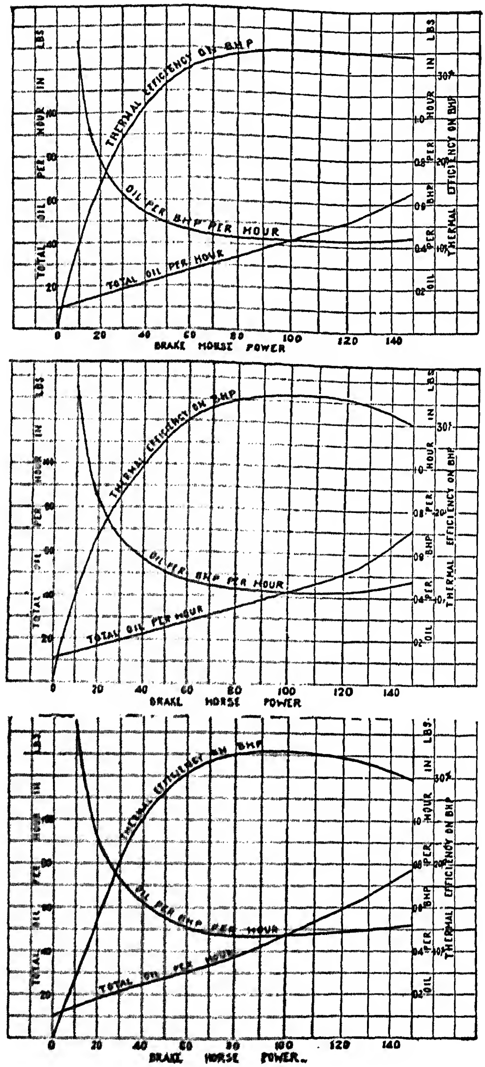

THERMAL EFFICIENCY ON BHP
OIL PER BHP PER HOUR
TOTAL OIL PER HOUR
TOTAL OIL PER HOUR IN LBS
OIL PER BHP PER HOUR IN LBS
THERMAL EFFICIENCY ON BHP
0 20 40 60 80 100 120 140
BRAKE HORSE POWER
THERMAL EFFICIENCY ON BHP
OIL PER BHP PER HOUR
TOTAL OIL PER HOUR
TOTAL OIL PER HOUR IN LBS
OIL PER BHP PER HOUR IN LBS
THERMAL EFFICIENCY ON BHP
0 20 40 60 80 100 120 140
BRAKE HORSE POWER
THERMAL EFFICIENCY ON BHP
OIL PER BHP PER HOUR
TOTAL OIL PER HOUR
TOTAL OIL PER HOUR IN LBS.
OIL PER BHP PER HOUR IN LBS.
THERMAL EFFICIENCY ON BHP
0 20 40 60 80 100 120 140
BRAKE HORSE POWER.

FIG. 52.

When this lever is in the position shown, the lower valve remains on its seat and a supply of lighter fuel is poured into the cup X. When the engine is started the pump will draw its supply from the cup. After a few revolutions the handle Y is moved over, the upper valve is then closed and the lower valve opened, enabling the pump to take its charges from the pipe Q which is connected directly to the oil heater V, shown in Fig. 48, through which the heavy oil is passed on its way from the oil tank to the pump so as to reduce its viscosity. The oil heater is attached to the exhaust pipe of the engine and the temperature of the heater can be varied by means of the large wing-valve shown. A strainer W is fitted in the oil heater. In case any vapor is formed a vent pipe Z is fitted at the top of the heater. A hand lever S is provided for charging the pump with oil before starting the engine. To stop the engine a small valve is opened in the oil delivery pipe between the oil pump and the oil sprayer. The engine is started by means of compressed air admitted through the valve shown at the back of the cylinder in Fig. 46.

The governing system has an instantaneous effect between the fuel and the power, and is accomplished by cutting off the supply of fuel at different points in the pump stroke according to the position of the governor and the load on the engine. The curves plotted in Fig. 52 show the fuel consumption, thermal efficiency, etc., of this engine and results obtained under test are tabulated in Table VII.

TABLE VII.—SUMMARY OF TESTS.*

CROSSLEY OIL ENGINE. 18½″×28″ STROKE
NORMAL RATED LOAD 117 B.H.P. AT 180 R.P.M.

B.H.P.	Oil per BHP per hr. Pounds	Oil per BHP per hr. Gram'es	Heat units per BHP per hr. B.Th.U. lbs. Fahr.	Heat units per BHP per hr. Calories Kg.-Cent.	Thermal Efficiency on the BHP Per cent
145.0 (24 per cent Over-load) Overload	0.432	196	7992	2014	31.84
123.0	0.427	194	7899	1992	32.22
96.0	0.424	192	7844	1975	3244
61.3	0.455	206	8417	2118	30.23
29.8	0.627	284	11599	2920	21.93
0	9.46 lbs. per hour	4300			
142.0 (21% Overload	0.457	207	8226	2073	30.93
125.0	0.425	193	7650	1928	33.26
96.7	0.424	192	7632	1923	33.35
30.9	0.647	293	11646	2935	21.85
0	11.0 lbs. per hour	5000			
146.0 (25% Overload	0.51†	233	8420	2122	30.22
129.0	0.504	228	8207	2068	31.01
102.0	0.475	216	7750	1953	32.84
67.0	0.488	222	8140	2051	31.26
32.0	0.700	318	11850	2986	21.47
.0	10.16 lbs. per hour	4610			

*Tests made by Prof. F. W. Burstall.

†These figures include a small proportion of Kerosene used for ignition purposes.

Fuels.		
Kerosene:	Calorific Value:	18,500 B.T.U. per pound
	" "	10,277 Calories per Kg.
	" "	18,000 B.T.U. per pound
Residual	" "	10,000 Calories per Kg.
Petroleum	" "	16,200 B.T.U. per pound
Tar Oil:	" "	9,000 Calories per Kg.

The De La Vergne "DH" horizontal hot surface crude oil engine is shown in section at Fig. 53. It is now built in sizes of 40 B.H.P. in one cylinder up to 130 B.H.P. in the twin cylinder design. Referring to Fig. 53 the main frame A is of the side girder construction and carries the main bearings for crankshaft. They are split at an angle of 30°, babbitted directly into the main casting. The cylinder casing is cast in one piece with the frame and is supported along its whole length by the main frame as shown. The cylinder liner B is held rigidly in place at its rear end, a rubber ring joint between it and the casing at the front end. The cylinder head D is a simple casting and to which are fitted the air inlet and exhaust valve housings, the vaporizer plate E and the oil inlet sprayer F inclined at an angle to allow the sprayer to impinge on this heated (uncooled) plate. The forged steel connecting rod C is fitted with adjustable bearings; that at the wristpin end being adjusted by the threaded bolt at the back end, the boxes being brought to the proper adjustment before the rod is put in position in the piston. The trunk piston L is made 1.8 to 2 times its diameter in length, thus the maximum pressure on the cylinder liner is approximately 20 lbs. per sq. in. The pressure on the wristpin bearing is below 1,800 lbs. per sq. in.,

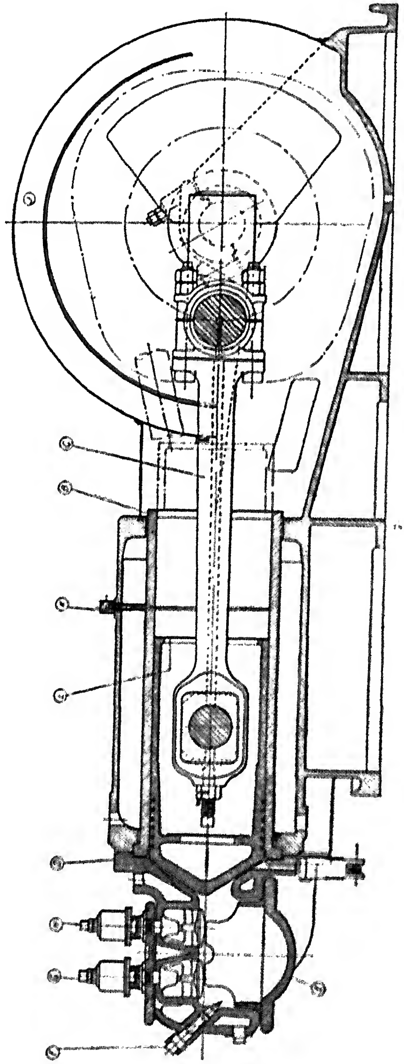


Fig. 53.

and that between the wristpin and piston lugs is below 3,000 lbs. The piston head is made as shown to conform to the combustion chamber shape. Six piston rings each ⅜″ width are placed ¾″ apart as shown on the piston. The crankpin splasher J is composed of an inner and outer section. The air for combustion is drawn in through the space between these sections. The air inlet valve G and the exhaust valve H are alike and are operated by the overhead levers actuated from the cams on the camshaft. Lubrication is furnished to all moving parts by force feed pumps, the lubricant entering the cylinder through the inlet shown at K.

The method of governing is the by-pass system which is illustrated by Fig. 11 and Fig. 12. As the governor balls C expand they force the sleeve E downwards against the left and right hand springs moving the lever F and fulcrum lever G. The two adjusting screws D impart motion to lever X which is in contact with the overflow oil valves K and Y. The former is the small pilot valve and the latter only opens when the governor is in its upper stroke. The fuel supply pump employed on this type is described and illustrated at Fig. 14. Indicator cards of this engine are shown at A, B and C on Fig. 58 and fuel consumption at various loads is recorded in Table VIII. The sprayer is described and illustrated at Fig. 4 and the sequence of valve movements is shown at Fig. 56.

De La Vergne "SI" (solid injection) type is shown in section at Fig. 54. This design is the most recent product of this builder. Its patented features are (*a*) the method of injecting the fuel which is sprayed into

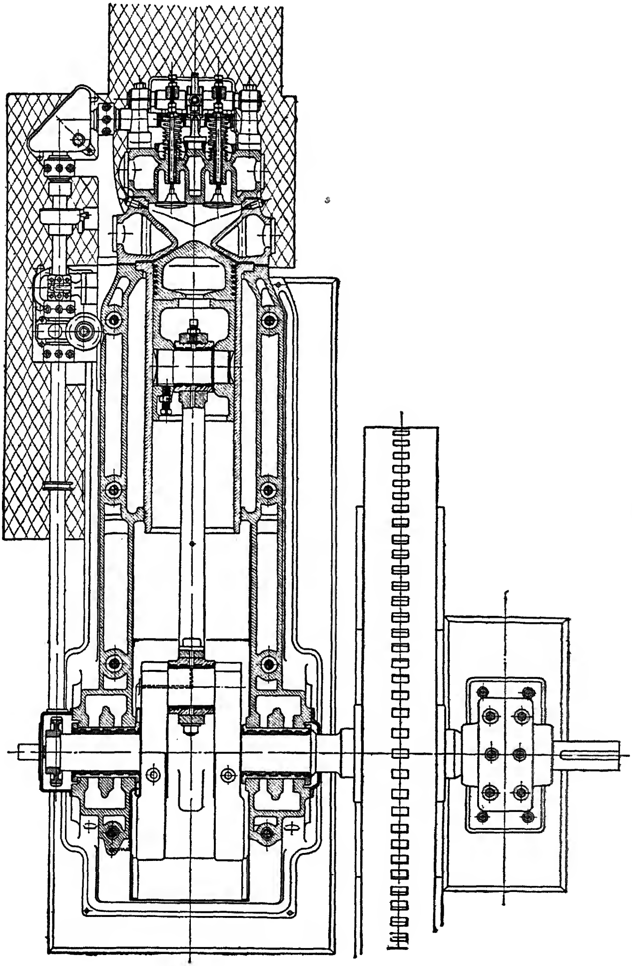


FIG. 54.

the combustion space at high pressure (about 2,000 lbs. per sq. in.) without air blast, (*b*) the use of two sprayers placed opposite each other at each side of the combustion space, (*c*) the specially designed and proportioned combustion space as shown in the illustra-

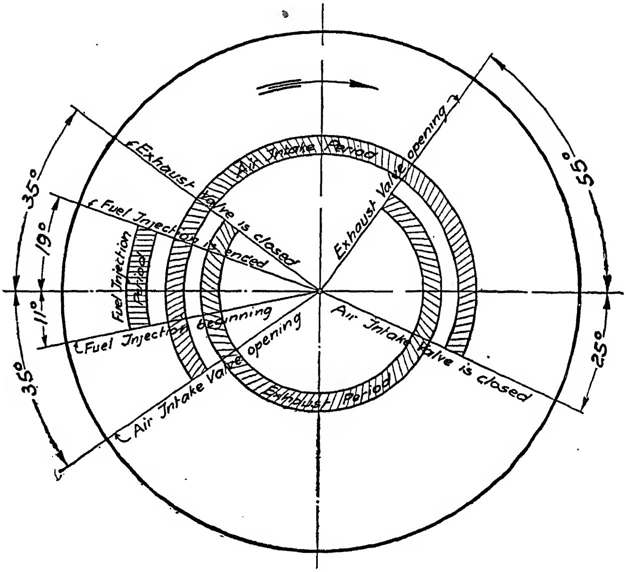


FIG. 55.

tion. The sprays of fuel in this design meet and impinge on each other in the center of the combustion space thus becoming thoroughly pulverized and intimately mixed with the air. The globules of oil vapor are thus well distributed throughout the whole of the

combustion space and the employment of a compression pressure of 350 lbs. is sufficient to raise the temperature to a point which will ignite (under all conditions) the mixture of intimately mingled vapor and air. No other

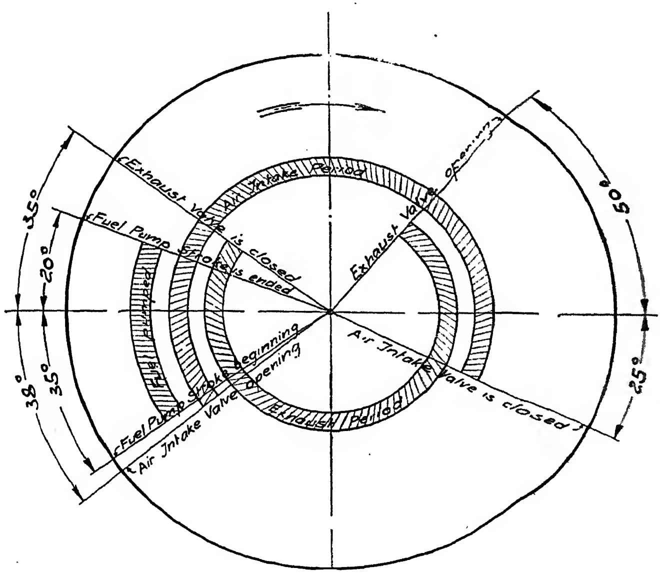


FIG. 56.

means of heat is necessary even in cold climates and the engine can be started satisfactorily in zero weather.

The spray valve is shown in section at Fig. 5. The sequence of valve movements is shown in Fig. 57 and the indicator diagrams from this engine are shown at D.E.F. at Fig. 58.

The method of governing with the "SI" type De La Vergne Engine is by means of a by-pass valve which is operated by the governor, but is arranged to open only during the latter part of the pump stroke; thus the fuel

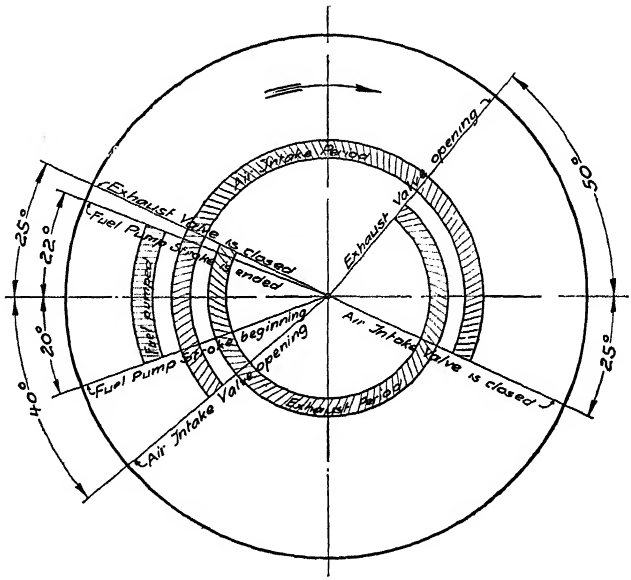


FIG. 57.

injection period begins at the same point under all conditions and loads, but the duration of the fuel injection period is varied and thus the correct amount of fuel is injected into the combustion space in accordance with the load on the engine. The air inlet and exhaust valves are placed in a horizontal position, each in

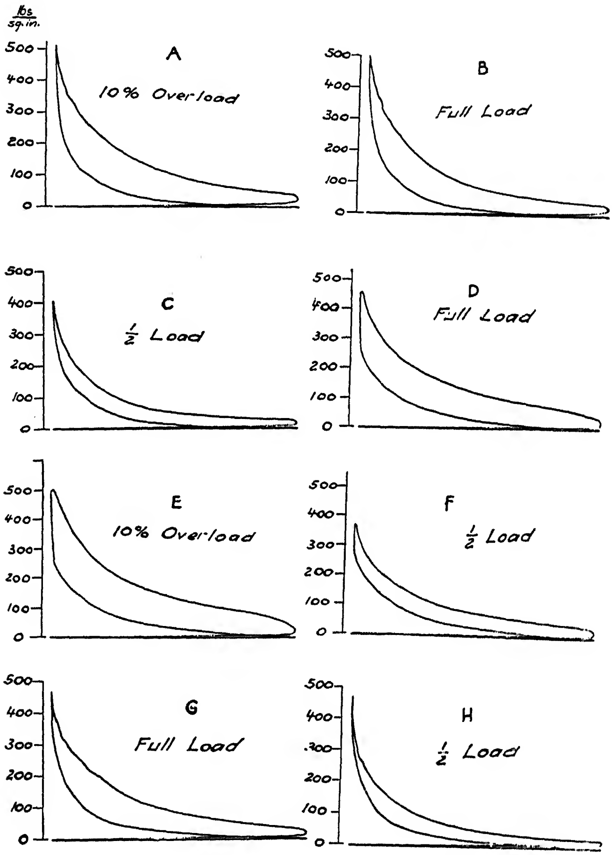

lbs / sq. in.
A
10% Over-load
B
Full Load
C
½ Load
D
Full Load
E
10% Over-load
F
½ Load
G
Full Load
H
½ Load
500
400
300
200
100
0

FIG. 58.

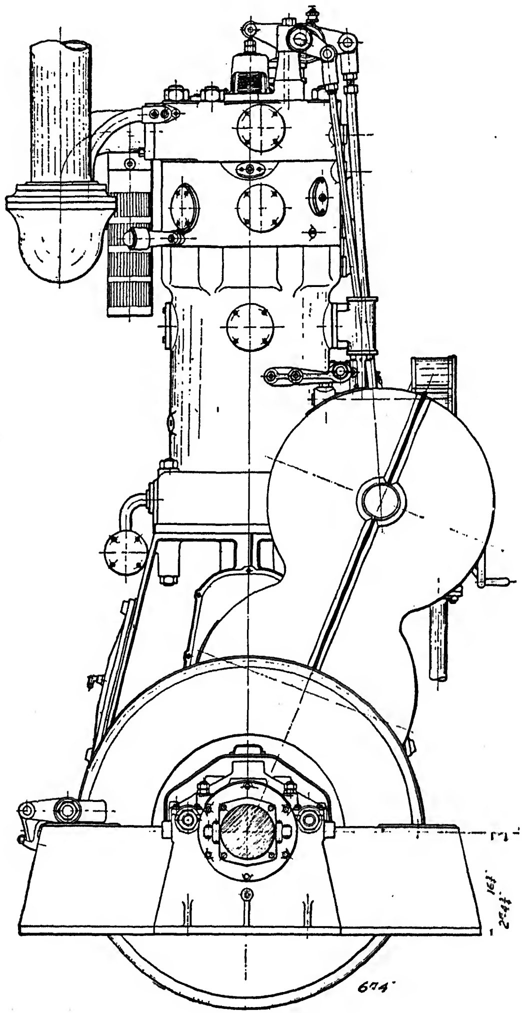


FIG. 59.

separate water-cooled valve-housing, properly guided, with sufficient bearing surface (see Fig. 54) in the guides to avoid undue wear. The starting air inlet valve is automatic on all types of engines made by this builder and is actuated by cam; thus the compressed air can only enter the combustion space during the correct period of the cycle as shown in Fig. 22.

De La Vergne "FH" type is referred to in Part II. The two-stage air compressor and compressor valves have been illustrated at Fig. 17 and Fig. 18. The valve movement diagram is shown at Fig. 55, and indicator diagrams have been reproduced at G & H in Fig. 58. The results obtained by the builders under test is shown in Table VIII in which are reproduced particulars of (*a*) the D.H. low compression type (*b*) the F.H. type having air blast fuel injection and (*c*) the S.I. solid injection type. In Figs. 59 and 60 are illustrated De La Vergne vertical S.I. type which operates on the same system of fuel injection as the horizontal solid injection type above referred to.

The Blackstone four-cycle oil engine is built at Stamford, England, in various sizes from 5.5 B.H.P. to 140 B.H.P. in a single cylinder. It is of heavy construction, the frame being carried back to the cylinder end, the separate cylinder liner being inserted from the back end and held in position at the front end with rubber ring, thus allowing for expansion lengthwise. There are various special features in this design, the fuel is injected by compressed air at the low pressure of approximately 450 lbs. furnished by the two-stage compressor placed at the side of the frame and actuated

TABLE VIII.—TESTS OF VARIOUS TYPES DE LA VERGNE OIL ENGINES.

Type	Rafted BHP	Cyl. diam. inches	Stroke inches	R.P.M.	Load	Fuel lbs. per BHP hr.	Compression press. lbs.	Max. press. lbs.	Duration Test	Outlet Water F.	Date Test
"D.H."	65	14	24	240	Full	0.485	195	465	1 hour	130°	Nov. 1, '20
"D.H."	65	14	24	240	10% over	0.502	195	465	1 hour	130°	
"D.H."	65	14	24	240	¾ Load	0.485	195	465	1 hour	130°	
"D.H."	65	14	24	240	½ Load	0.53	195	465	1 hour	130°	
"F.H."	140	20	34½	164	Full	0.48	275	475	1 hour	180°	Nov. 27, '19
"F.H."	140	20	34½	164	10% over	0.483	275	475	1 hour	180°	
"F.H."	140	20	34½	164	¾ Load	0.468	275	475	1 hour	180°	
"F.H."	140	20	34½	164	½ Load	0.5	275	475	1 hour	180°	
Twin S I	200	17	27½	200	Full	0.420	320	540	1 hour	150°	Oct. 5, '20
S I	200	17	27½	200	10% over	0.421	320	540	1 hour	150°	
S I	200	17	27½	200	¾ Load	0.410	320	540	1 hour	150°	
S I	200	17	27½	200	½ Load	0.458	320	540	1 hour	150°	

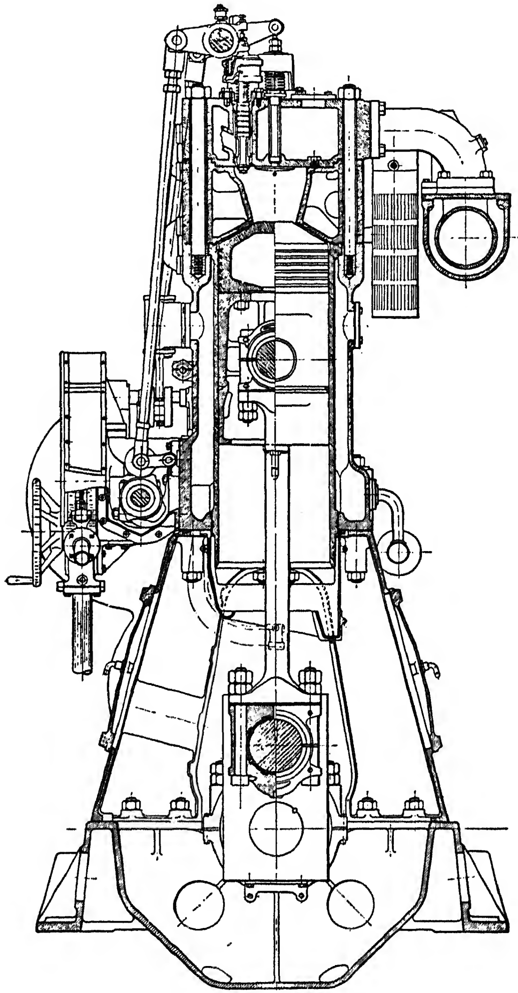


FIG. 60.

by eccentric attached to the end of the crankshaft.

The compressed air is conducted from the compressor to the air container and thence to the spray valve box also to the combustion space, the latter connection is for starting purposes. The spray valve box with its two systems of sprayers, the hot bulb or vaporizer, the two streams of fuel vapor entering the combustion space and the deflecting plate placed at the back of the piston as shown in detail at Fig. 61. The method of operation is as follows:

The piston moves outwards taking in air supply through the air inlet valve placed in its separate housing or cage above the combustion space. The fuel is supplied by the fuel pump situated at the forward end of the engine actuated from crankpin in crankshaft and rocker arm, the amount of fuel delivered being varied by the wedge arrangement controlled from the governor placed near the crankshaft main bearing and which lengthens or shortens the stroke of the pump in accordance with the load on the engine. The fuel is then conducted to the spray valve box. A feature which is peculiar to the Blackstone design is the use of two separate sprays of fuel. The one spray injects fuel at all loads directly into the hot bulb marked I, Fig. 61, and which at very light loads is sufficient vapor when ignited to develop the required pressure and it also serves to maintain this chamber at sufficient temperature and no "cooling off" is experienced when operating at no load. The second spray is injected into the cylinder or combustion space direct where it is ignited from the burning vapor issuing from the hot chamber,

which receives its vapor under all operating conditions from the other spray valve as previously referred to.

The mechanism in the spray valve box (Fig. 61) consists of the main spray valve A injecting directly

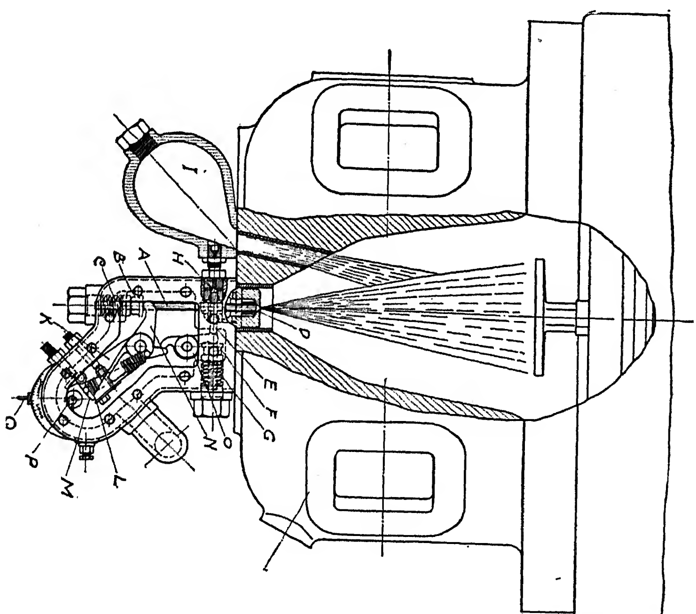


Fig. 61.

into combustion space held on its seat D by the spring C and plug B. The ignition spray valve injecting into the hot bulb I is shown at E placed at right angles to

valve A. It is held on its seat by spring G and nut F. The spray valves are actuated by a rocker arm operated from a pin placed eccentrically in the end of the camshaft; this rocker arm causes the spindle K to move to and fro in the spray valve box. As the air pressure used for injection of the fuel is admitted to the interior of the spray valve box, it is necessary that the spindle K (the only moving part where leakage of air could take place), be kept air tight which is effected by a fibre washer pressed against the end of the bushing in which it rotates. Attached to the inner end of the spindle K is an arm which carries the hardened trip paul M. When this part comes in contact with the hardened steel lever L, it moves the main spray valve lever N and allows this spray valve to open, and also by means of the rocker arm O, it also opens the ignition spray valve. This motion continues until the end of lever N hits the eccentric pin P, which allows the end of the paul to disengage from under the lever L. Then the springs return each valve to their seats. The position of the pin P is adjusted by the handle Q so that the air pressure in the air container is maintained at 450 lbs. per sq. in. As the piston returns on its compression stroke when compression is nearly completed the spray valves are opened by the movement of the rocker arm and the compressed air injects the fuel into the hot bulb and also into the combustion space in separate sprays. Ignition begins in the hot bulb and combustion follows in the cylinder which takes place at constant volume, but as will be seen from the indicator diagrams in Fig. 62 it appears to be slow in

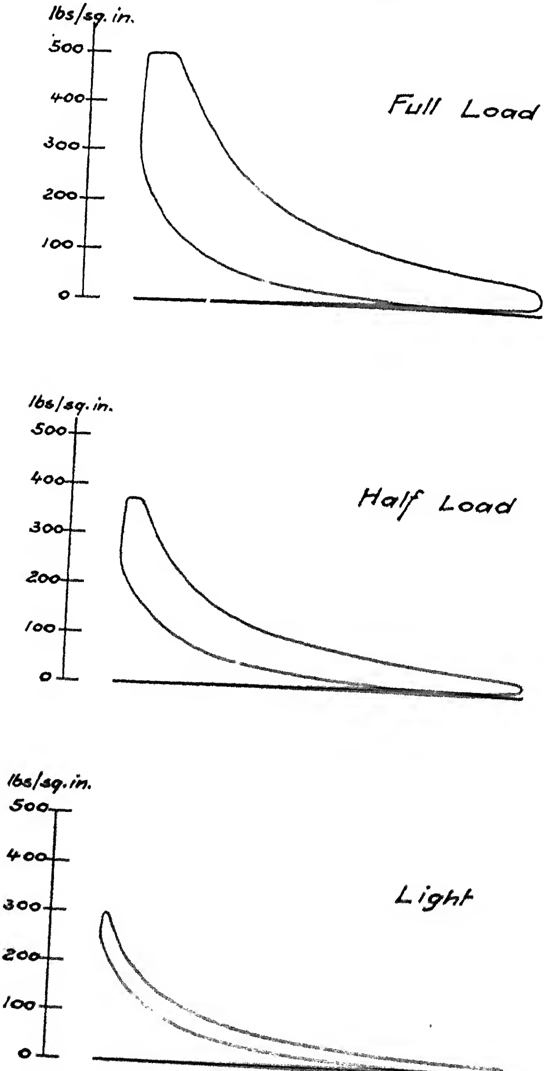

lbs/sq. in.
500
400
300
200
100
0
Full Load
lbs/sq. in.
500
400
300
200
100
0
Half Load
lbs/sq. in.
500
400
300
200
100
0
Light

FIG. 62.

completion, as the pressure is constant during the beginning of the outward stroke of the piston. This type has been described as a "Dual combustion" engine because of this feature.

In one size the following dimensions, etc., were recorded:*

Cylinder diameter 15.5", stroke 23" revolutions per minute 205, compression pressure 150 lbs. per sq. in., maximum pressure 300 lbs., injection air pressure 450 lbs., mean effective pressure 80 lbs., air compressor first stage 6½" diameter, second stage 2⅞", stroke 5", mechanical efficiency 80%.

Fuel consumption, full load, 0.465 lbs.
Three-quarter load, 0.495 lbs.
Half load, 0.55 lbs.
Fuel: 27° Baumé (0.896 specific gravity).
Heat value 19,300 B.t.u.

THE HORNSBY TYPE "R" four-cycle oil engine which is one of the latest designs is shown in Figs. 63 and 64. The earlier designs of the "Hornsby-Akroyd" oil engine is described in Part II of this treatise. The modern Hornsby oil engine preserves the simplicity of design which made the former engines so famous and has the added advantage of greater economy. This improvement has been achieved in great measure by (*a*) allowing increased surfaces of the combustion chamber or vaporizer to be exposed to the cooling effect of the circulating water spaces surrounding them, (*b*) to increased compression pressures which

*"The Engineer," April 23rd, 1915.

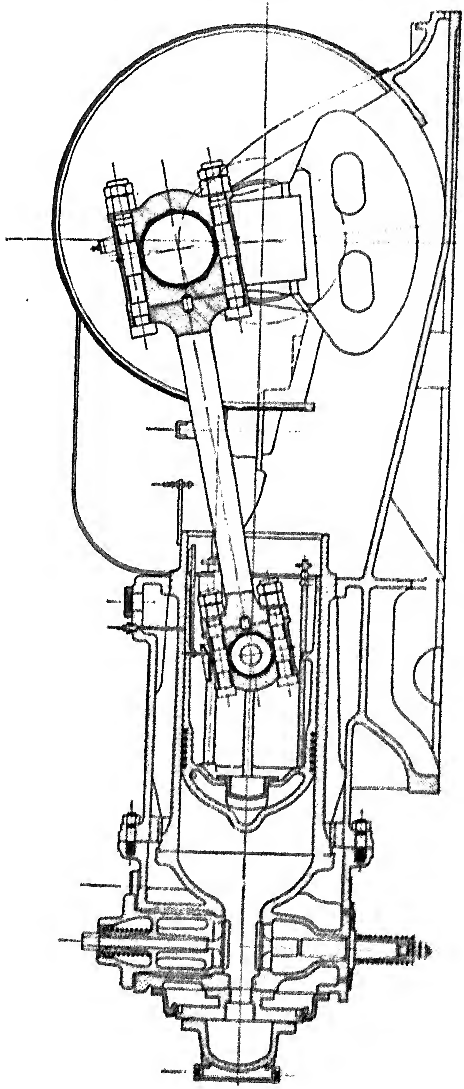


Fig. 63.

can be observed by comparing the indicator cards reproduced with the former design with those illustrated from the modern engine at Fig. 65. The present compression pressure is approximately 260 lbs. The fuel is now injected as the piston is nearing the end of the

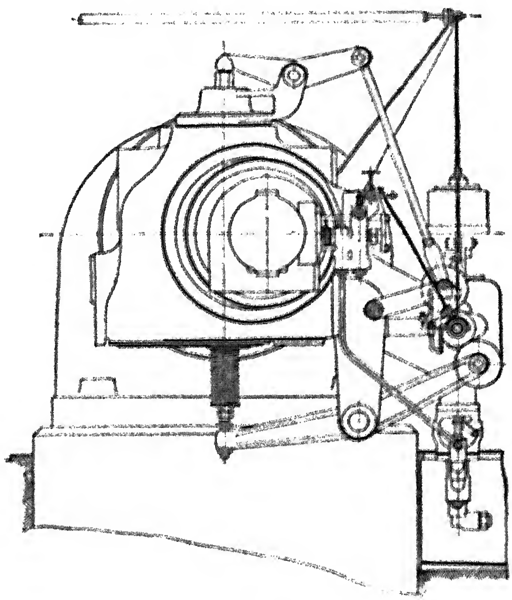


FIG. 64.

compression stroke and thus air only is allowed in the cylinder and combustion space during the process of compression. The valve motion, cams and governor are shown in Fig. 64. The fuel pump is operated from

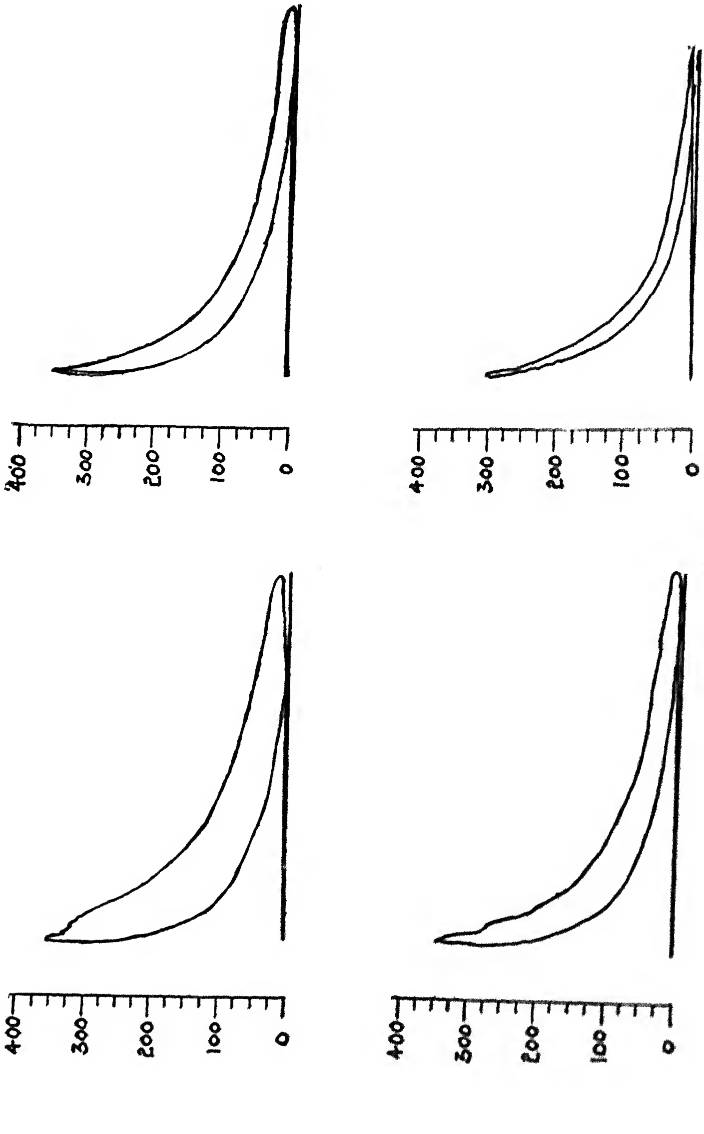

400
300
200
100
0
400
300
200
100
0
400
300
200
100
0
400
300
200
100
0

FIG. 63.

a separate cam instead of from that operating the air inlet valve as was arranged in the earlier engines.

The air inlet and exhaust cams are made of cast iron unchilled but of such width that the pressure between the rollers and cams is distributed over sufficiently

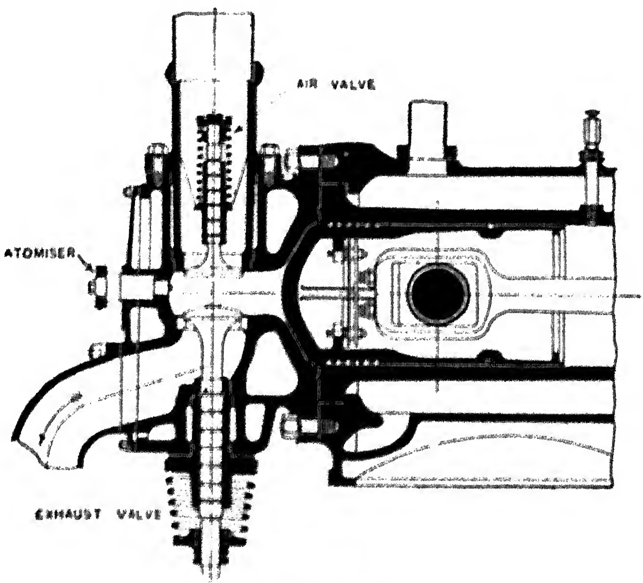


FIG. 66.

large surfaces and is so reduced as to make surface hardening unnecessary. In addition to these cams, a governor cam is also employed placed on the end of the camshaft shown in Fig. 64; this cam slides on a spiral key in the camshaft, its position being directly regulated by the governor. Regulation of speed is effected

by by-passing the fuel through the overflow valve shown in Fig. 6. The vertical relief valve is mechanically operated from the cam, controlled by the governor. In Fig. 6 is also shown the horizontal fuel inlet valve held against its seat by a spring and the back pressure ball valve.

The fuel consumption per B.H.P. hour is shown by the following tests:

TABLE IX.—320 B.H.P. HORNSBY TYPE "R" OIL ENGINE TEST.

DIRECT-CONNECTED TO 200 K.W. D.C. GENERATOR
March, 1916.

Oil:	Texas Fuel Specific Gravity .900 at 60° Fahr. (26° B). Flash Point about 230° Fahr. (Open Test).	
Average B. H.P.	Average R. P. M.	Lbs. of Fuel per B.H.P. per hour
306.41	170.00	.47
231.54	170.48	.48
151.33	170.86	.55
78.27	171.22	.75

GOVERNOR TRIALS.

Carrying full Load, the engine speed was	170 R.P.M.
Immediately the Load was suddenly taken off the speed was...........	175 " " "
and it settled down to................	171 " " "
Immediately the Load was suddenly put on the speed was................	165 " " "
and it settled down to................	170 " " "

OVERLOAD.—Engine then ran ½ hour at 10% overload, followed for ½ hour at 20% overload.

TABLE X.—85 B.H.P. HORNSBY TYPE "R" OIL ENGINE TEST.

February, 1919.

CONSUMPTION IN LBS. PER BHP PER HOUR.

Name of Oil	Beaumé	Spec. Grav.	BHP	Consump.	BHP	Consump.	BHP	Consump.
Royal Daylight	40°	.800	93	.47	85	.48	42	.54
Solar Oil	30°	.880	89	.49	82	.47	40	.56
Texas Fuel Oil	27	.897	84	.47	77	.46	37	.56

Indicator diagrams taken at full load, three-quarter load, half and light load are reproduced in Fig. 65.

THE RUSTON four-cycle horizontal crude oil engine, built by Ruston & Hornsby, Lincoln, England, is shown in section at Fig. 66.* The compression pressure is 430 lbs. per sq. in., M.E.P. 85 lbs. The fuel economy as low as .4 lbs. per B.H.P. hour has been obtained using heavy fuel or crude oils. With this compression pressure starting "cold" is readily performed. The construction of the cylinder head, air inlet and exhaust valves, and the atomizer or fuel injection valve is shown in Fig. 66. This type is built in sizes up to 170 B.H.P. in one cylinder. The curves in Fig. 67 show the fuel consumption at varying loads, and in Fig. 68 the curves show a comparison of the results obtained with this engine with those of the average Diesel type.

THE CAMPBELL high compression four-cycle modern oil engine manufactured at Halifax, England, is

*Figs. 66, 67 and 68 are reproduced from paper presented by Mr. F. H. Livens before *Inst. Mech. Engrs*: *London*, 1920.

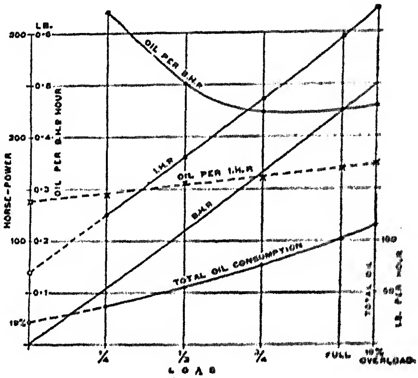

LB.
300
0·6
0·5
200
0·4
0·3
100
0·2
0·1
10%
HORSE-POWER
OIL PER B.H.P HOUR
OIL PER B.H.P
I.H.P
OIL PER I.H.P
B.H.P
TOTAL OIL CONSUMPTION
100
50
TOTAL OIL
LB. PER HOUR
¼
½
¾
FULL
10%
OVERLOAD
LOAD

FIG. 67.

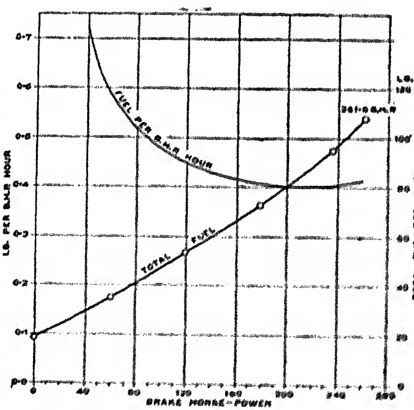

0·7
0·6
0·5
0·4
0·3
0·2
0·1
0·0
LB. PER B.H.P HOUR
FUEL PER B.H.P HOUR
TOTAL FUEL
LB.
120
100
80
60
40
20
0
TOTAL FUEL PER HOUR
0
40
80
120
160
200
240
280
BRAKE HORSE-POWER

FIG. 68.

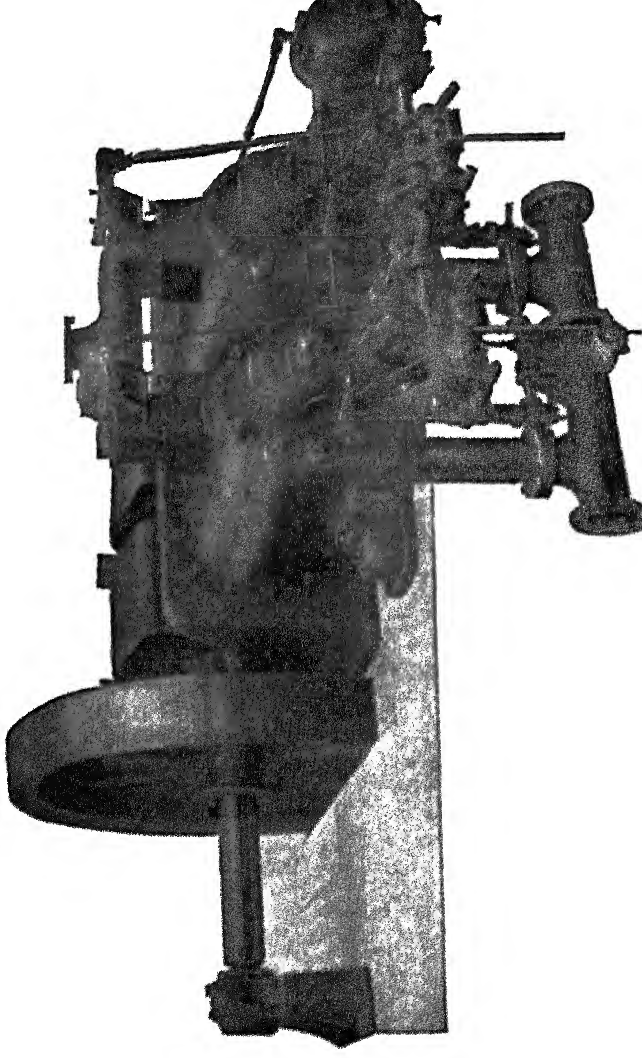
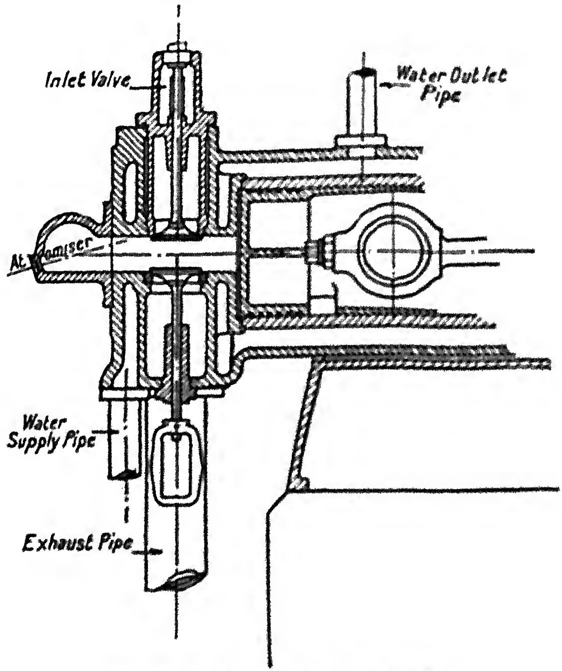


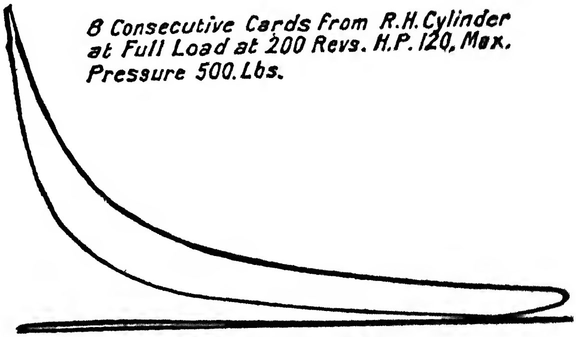
Fig. 69.

shown at Fig. 69. It is built in various sizes from 7 to 100 B.H.P. one-cylinder design and in multicylinder type to 400 B.H.P. The illustration shows a twin cylinder 120 B.H.P. engine with cylinders 15" diameter and 24" stroke operating at 200 revolutions per minute. In the sectional view of the back end of the cylinder seen in Fig. 70 the air inlet valve with separate housing or cage placed above, and the exhaust valve placed beneath it are shown. The compression pressure in the cylinder before ignition is approximately 360 lbs. per sq. in. The maximum pressure as shown on the full load indicator card reproduced in Fig. 71 is 500 lbs. and the M.E.P. is about 85 lbs. The maximum pressure decreases to 450 lbs. at three-quarter load and to 400 lbs. as shown in the no load indicator card. This engine starts cold without heating lamp. The fuel is injected into the chamber shown in Fig. 70 placed at the back end of the cylinder concentrically with the cylinder. The liquid fuel is injected under very high pressure created by the action of the pump against a needle valve held on its seat by strong spring. The fuel enters at the end of the compression stroke, ignition and fuel injection being practically simultaneous. The fuel pump is operated by a cam placed on the camshaft which is actuated by steel and bronze gears from the crankshaft. Combustion takes place at constant volume and not at constant pressure as in the true Diesel engine. The fuel is injected into the vaporizing chamber in a solid state, that is without air blast. The following test (Table XI) was made with Mexican fuel oil of the characteristics indicated

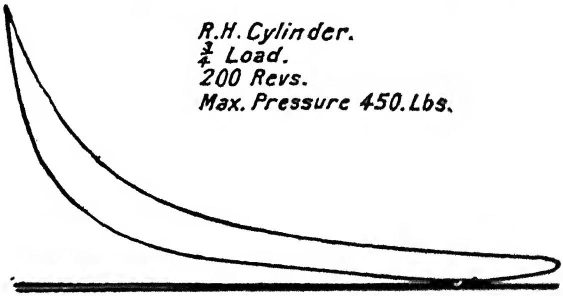


BREECH END OF CYLINDER.

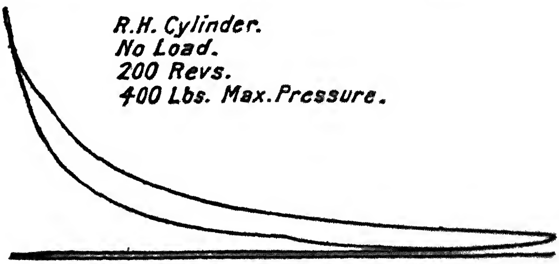
FIG. 70.



FULL POWER CARD.



THREE-QUARTER POWER CARD.



NO LOAD CARD.

FIG. 71.

when the fuel consumption was approximately 0.5 lbs. per B.H.P. per hour. A slight amount of light fuel oil is used when starting and stopping so as to prevent the heavier fuel congealing in the oil pipes when the engine is standing cold.

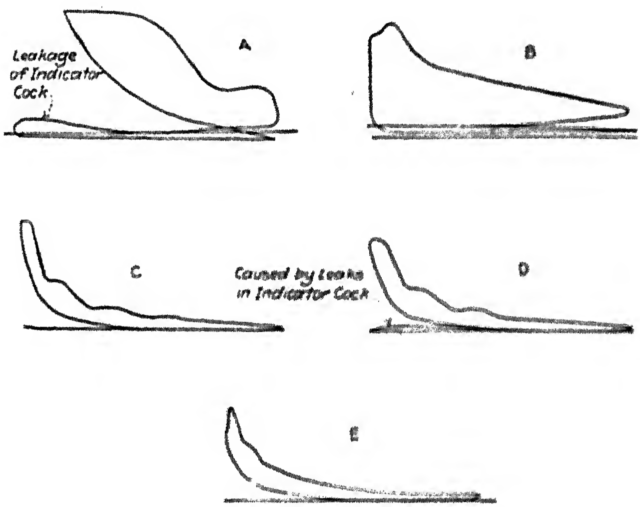


FIG. 72.

Governing is effected by a wedge placed between the pump plunger and actuating rocker controlled by the governor and which lengthens or shortens the stroke of the pump as required. When using heavy fuel such as that named below, it is heated by passing through a chamber which has its outer jacket heated from the exhaust gases.

TABLE XI.—TEST ON CAMPBELL OIL ENGINE WITH MEXICAN FUEL OIL.

Duration of test. hours	Average BHP Developed	RPM	Fuel per BHP. Hr. Lbs.	Cooling Water Wght per Min.	Inlet Temp. Fahr.	Outlet Temp. Fahr.
12	31.53	223	0.54	14.0	54.0	149.0
24	31.8	226	0.498	14.0	52.0	154.0
30	31.8	226	0.514	14.0	52.5	150.0
30	24.6	230	0.482	10.0	59.0	159.0
30	16.75	233	0.51	7.0	59.5	150.0
30	8.5	240	0.656	4.0	58.0	152.0
1	35.0	220	0.56	14.0	58.0	152.0

FUEL.

Spec. gravity 60 deg. F.	0.964
Beaumé	15 deg.
Flash Point (close test) F.	244 deg.
Calorific value gross per lb.	18048 BTU
Calorific value net per lb.	17890 BTU
Viscosity 100 deg. F. (Redwood #1)	1500 seconds

Analysis:

Carbon	83.52%
Hydrogen	11.68%
Sulphur	3.27%
Ash	0.16%
Undetermined	1.37%

THE HVID (Brons, Burnoil) engine now being manufactured in various sizes from 1½ B.H.P. upwards is of the four-cycle type and is built by various manufacturers both in the vertical and horizontal design. The compression pressure before ignition is carried to between 425 and 475 lbs. per sq. in., thus the heat developed by that process is approximately 900° to 1,000° F. A typical indicator diagram from this engine is shown at Fig. 72. In the cylinder head is placed the fuel admission valve through which the liquid fuel is admitted to a small steel cup and is shown in section at Fig. 73. During the outward or suction stroke of the piston air only is drawn into the cylinder through the main air inlet valve and simultaneously, fuel is drawn or gravitates to the small cup through the pin hole which is uncovered as the fuel valve is lifted from its seat. The amount of fuel thus entering this receptacle is regulated by the needle valve

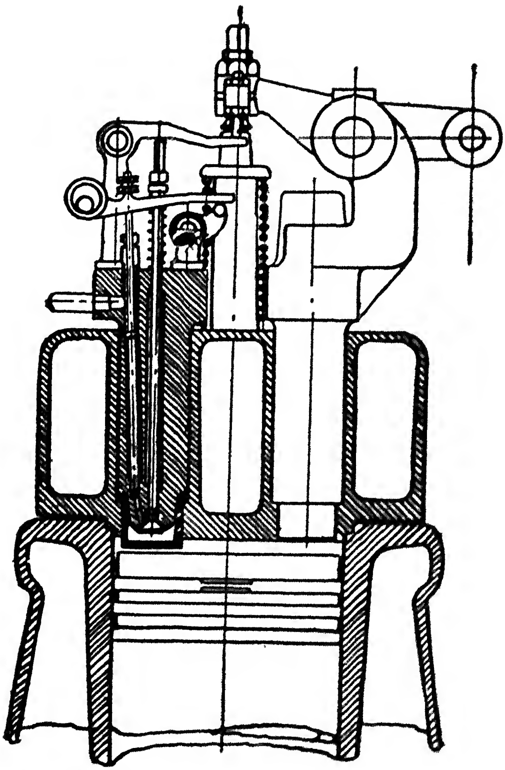


FIG. 73.

which in turn is controlled by the governor. During this period a slight amount of air is allowed to enter the cup through the auxiliary air passage. When the compression stroke commences all valves are closed and the heated air towards the end of the compression

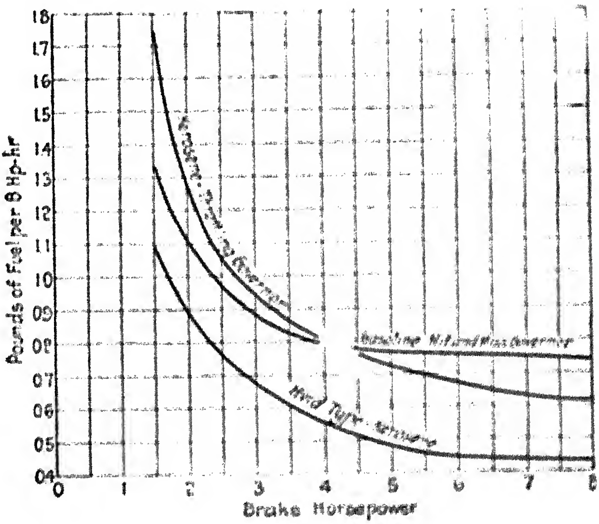


FIG. 74.

period rushes through the small holes in the sides of the fuel cup which vaporizes the fuel it contains. Ignition follows, starting in the cup, which action quickly ejects the fuel or vapor from the cup into the combustion space where ignition and rapid combustion of all vapor then takes place. Fig. 72 shows a series of

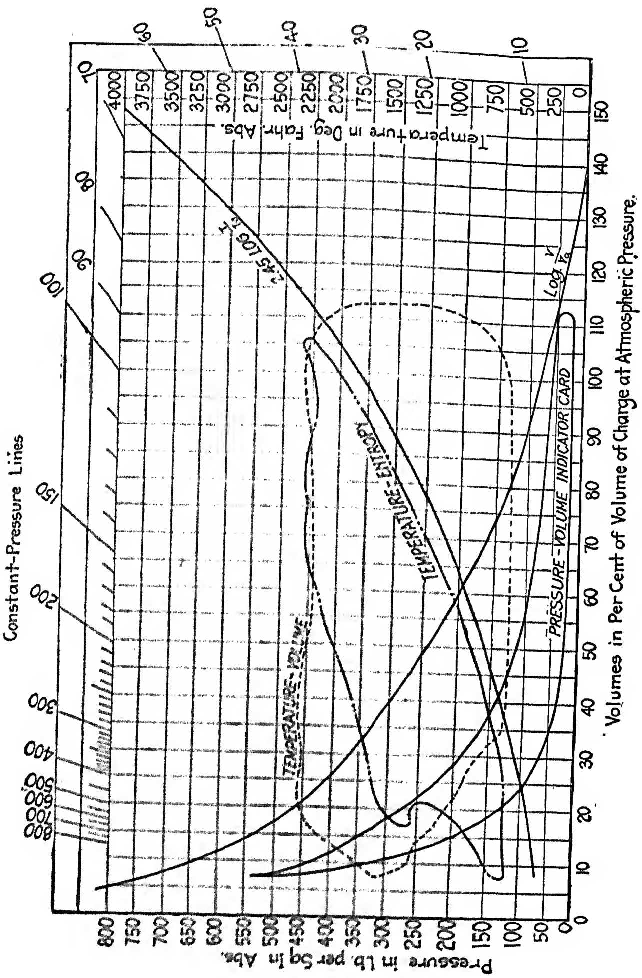

Constant-Pressure Lines
Temperature in Deg. Fahr. Abs.
Volumes in Per Cent of Volume of Charge at Atmospheric Pressure.
Pressure in Lb. per Sq. In. Abs.
TEMPERATURE-ENTROPY
TEMPERATURE-VOLUME
PRESSURE-VOLUME INDICATOR CARD

FIG. 75.

indicator diagrams taken from an 8 H.P. and 20 H.P. engine of this design.*

Diagram A is taken with a light spring. The air suction line is seen below the atmospheric line and is thus allowed so as to remove the residual pressure remaining in the cup and assist the fuel inlet of the following cycle. Diagram B shows the pressures that exist in the cup itself from which the vapor enters the combustion space. Diagram C was taken at full load and that at D with 25% overload. The diagram E is from a 20 H.P. (8½" x 10") at full load.

In Fig. 74 the different curves indicate the comparative fuel economy of the Hvid engine and other engines of similar capacity and speed.

In Fig. 75 is shown the entropy diagram plotted from a pressure volume indicator diagram taken from a 5¾" x 9" single cylinder Hvid oil engine operating at 450 R.P.M. using ordinary Kerosene as fuel and reproduced together with Figs. 73 to 77 from E. B. Blakely's paper referred to.

The heat balance and efficiency curves shown in Figs. 76 and 77 were recorded with an engine 5¾" diameter and 9" stroke when flexibly connected to a Sprague electric dynamometer. The fuel (Kerosene) consumed in these tests contained 19.740 B.t.u. per lb.

*Reproduced by permission from the paper presented by E. B. Blakely before the Amer. Soc. of Mech. Eng., Dec., 1919.

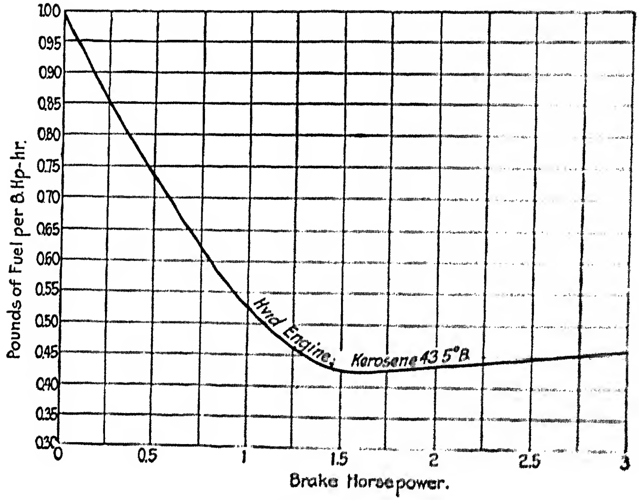

Pounds of Fuel per B.Hp-hr.
Hvid Engine, Kerosene 43.5°B.
Brake Horsepower.

FIG. 76.

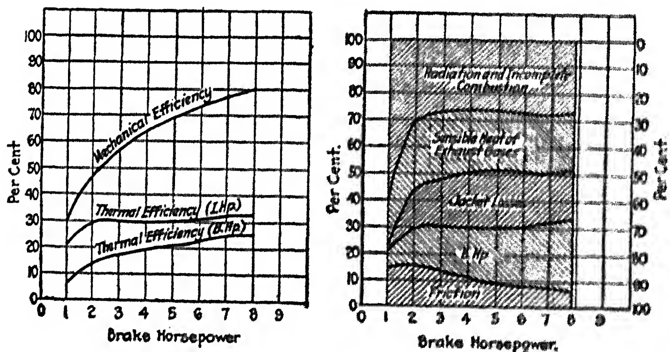

Per Cent
Mechanical Efficiency
Thermal Efficiency (I.Hp.)
Thermal Efficiency (B.Hp.)
Brake Horsepower
Per Cent.
Radiation and Incomplete Combustion
Sensible Heat of Exhaust Gases
Jacket Losses
B.Hp.
Friction
Per Cent
Brake Horsepower.

FIG. 77

INDEX. PART I

	PAGE
ADIABATIC expansion....	10
Adiabatic line	8
Advantages of cycles....	12
Air brake injection	5
Air compressors	29
Air compressors, lubrication	62
Air inlet pipes	53
Air inlet valve 29,	108
Air standard diagram...	9
Air supply	59
Airless injection........	7
Am. Soc. Mech. Engineers' Code of Tests..	42
BALANCING reciprocating parts	40
Beardmore two-cycle type	67
Bessemer engine........	82
Blackstone engine	111
Blackstone test	112
Bolts, connecting rod....	27
Bolts, cylinder head	18
Buckeye engine	84
CAMPBELL engine........	124
Campbell test	131
Cams	122
Capacities of engines ..	6
Carbon deposits	72
Carnot cycle	9
Clerk, Sir Dugald......	9
Combustion space, 18, 84, 114,	118
Compression cycles......	9
Compression pressure, 3, 5, 78, 96, 107, 118, 124 127,	132
Connecting rods 24,	102
Connecting rod bolt	27
Cooling arrangements.54,	82
Cooling system, enclosed.	54
Cooling water, amount ..	56
Cooling water, instructions	57
Correction of engine.....	57
Crank case compression, 58,	74
Crank shaft	32
Crank shaft specifications	37
Crosshead	85
Crossley oil engine.....5,	92
Crossley test	101
Cycles of operation.....	12
Cylinder casing	17
Cylinder design	15
Cylinder dimension, 78, 88, 118, 127,	136
Cylinder head bolts......	18
Cylinder liner	16
Cylinder lubricants	61
DEFICIENT power, causes of	60
De La Vergne "D H" engine	102
De La Vergne "S I" engine	104
De La Vergne "F H" engine	111
Diesel patents 1,	5
Diesel type comparison..	124
Dimensions, crankshaft..	35
Dual combustion	118
EFFICIENCIES, various.13,	48
Efficiency of engine.....	11
Efficiency, volumetric ...	13

PAGE
Engine, Bessemer 82
Engine, Blackstone 111
Engine, Buckeye 84
Engine, Campbell 124
Engine, Crossley 5, 92
Engine, De La Vergne "D H" 102
Engine, De La Vergne "S I" 104
Engine, De La Vergne "F H" 111
Engine, Fairbanks Morse 80
Engine, Hornsby type R. 118
Engine, Hvid 132
Engine, Petter 74
Engine, Ruston 124
Engine, Worthington ... 86
Entropy diagram 136
Exhaust gases, analysis.. 47
Exhaust pipes 52
Exhaust ports 31
Exhaust valve 29, 108
Exhaust velocity 72

FAIRBANKS Morse Marine Engine 80
Flywheels 40
Foundation for engine. 52, 92
Four-cycle type 90
Frame, main 15
Fuel analysis ...102, 118, 132
Fuel consumption, 1, 70, 84, 86, 124
Fuel economy 88
Fuel injection period, 5, 70, 120, 127
Fuel injection, variable.. 74
Fuel pump 80, 104
Fuel supply 58

GOVERNING engine.....98, 100, 104, 108, 122, 123, 130

HEAT balance 43, 136
Heat losses 11

PAGE
Heater, heavy oils...... 100
Heating lamp 85
Horizontal design 91
Hornsby type "R" engine 118
Hornsby type "R" engine test 123
Hot surface type oil engines 1
Hvid engine 132

IGNITION 3, 116, 134
Ignition, modern engines. 6
Ignition, requirements of 58
Ignition, timing of...... 59
Indicator card, 60, 68, 107, 117, 129, 136
Indicator card, low pressure 59
Injection air pressure.... 111
Installation of engine.... 50
Isothermal line 8

KNOCKING in engine..... 59

LEAKAGE, air inlet and exhaust valves 61
Liner, cylinder 16
Lloyds rules, crankshafts 38
Lubricants 61, 88, 104
Lubrication, air compressor 62

MAIN frame 15
Marine engine, Fairbanks More 80
Marine engine, Petter... 80
Mean effective pressure, 70, 124, 127
Mechanical efficiency.... 118

OIL pump 24, 96
Oil spray 59
Operation of engine.... 57
Over load capacity...... 90

PAGE

PARALLEL operation...... 41
Petter test 79
Petter two-cycle engine.. 74
Piston blowing 60
Piston lubricants 61
Piston rings 104
Piston speeds 40, 72
Pistons 31, 102

POWER deficient, causes of 60
Pre-ignition 3
Pressure of compression. 3
Pressure of piston 102
Pressures, two-cycle 68
Pulverizing devices...... 7
Pump, oil 24

RANKINE cycle 9
Ratio stroke and diam.. 40
Reciprocating parts, balancing 40
Regulation of speed..21, 98, 100, 104, 108, 122, 123, 130
Reversial, direction rotation 50
Reversing engine 80
Ruston engine 124

SCALE in cylinder head.. 56
Scavenging air..72, 85, 86, 96
Silencer, exhaust 57
Solid injection 7, 106
Specific heat 8
Speed, piston 40
Speed, regulation of..21, 98, 100, 104, 108, 122, 123, 130
Spray, oil 59
Sprayer, friction of..... 21
Spraying devices....7, 18, 82, 85, 88, 96, 106, 114, 132

PAGE

Starting 5
Stroke, ratio 40
Supply, fuel 58

TAR oils 72
Temperature, hot surface 70
Temperature of compression 132
Temperature, outlet water 56
Test, Blackstone 112
Test, Campbell 131
Test, Crossley 101
Test, Duration of 42
Test, Hornsby "R" 123
Test, Petter 79
Test, results of 46
Testing 42
Thermal efficiency 11
Thermodynamics 7
Thrust bearing 82
Turbulence 96
Two-cycle type 63
Two-cycle, various types. 66

VALVE housing 111
Valve settings 50, 106
Valves 29
Valves, air and exhaust, leakage of 61
Vaporizer 84
Vapor mixtures....8, 58, 59
Vertical design 91
Volume, constant........ 127
Volumetric efficiency..13, 72

WEIGHT of engine...... 86
Worthington engine 86
Wrist-pin pressure...... 102

PART II

Historical and Earlier Types

OF

LOW COMPRESSION OIL ENGINES

CONTENTS. PART II

CHAPTER I.

INTRODUCTORY.

PAGE

Historical—Classification of Oil Engines—Various Vaporizers—Different Igniting and Spraying Devices—The Different Cycles of Valve Movements 1-19

CHAPTER II.

ON DESIGNING OIL ENGINES.

Simplicity in Construction and Arrangement of Parts—Comparison of Oil and Gas Engines—Cylinders, Different Types—Cylinder Clearance—Crank-shaft, Dimensions and Formulæ—Balancing of Crank-shafts Described—Connecting-rods, Strengths, etc.—Piston, Piston-rings—Piston speed—Fly-wheels, Formula for—Air and Exhaust Cams—Cylinder Lubricators—Valves and Valve-boxes—Velocity of Air through Valves—Crank-shaft Bearings—Proportions of Engine Frame—Crank-pin Dimensions—Valve Mechanisms, Gearing and Levers—Governing Devices—Exhaust Bends—Oil-supply Pump—Oil-tank and Filter—Comparison of Horizontal and Vertical Type Engines, with Advantages of Each—Two-cylinder Engines Discussed—Assembling of Oil-engines—Scraping in Bearings—Fitting of Piston and Piston-rings—Fitting Connecting-rod Bearings—Fitting Air and Exhaust Valves—Testing Water-jackets—Fly-wheel Keys—Oil-supply Pipes—Cylinder Made in Two or More Parts, .. 20-58

CHAPTER III.

TESTING ENGINES.

PAGE

Object of Testing—Comparison with Steam-engines—Different Records to be Taken—Diagram for setting Valves—Preparing for Test—Heating of Vaporizer—Starting—Difficulties of Starting—Compression, How to Test—Leakage of Valves and Cylinder—Lubrication of Piston and Bearings—Easing Piston—Synonymous Terms for Power Developed—Indicated Horse-power—Brake, Horse-power—Indicator—Fully Described—Reducing Motions—Planimeters—Indicator-cards described in Detail and Analyzed—Defects as Shown by Indicator—How to Remedy Same—Early and Late Ignition, How to Alter—The Compression and Expansion Line—Choked Exhaust—Mean Effective Pressure, How to Increase—Back Pressure of Exhaust—Tachometers—Fuel-consumption Test Fully Described—Mechanical Efficiency—Thermal Efficiency—Table of Disposition of Heat—Valve Diagram—Exhaust Gases—Complete and Incomplete Combustion—Testing the Flash-point of Kerosene—Viscosimeter, ... 59-95

CHAPTER IV.

COOLING WATER-TANKS AND OTHER DETAILS.

Water Connections—Capacity of Tanks Required—Gravitation System of Circulation—Water-pumps—Connection to City Water Main—Temperature of Outlet Water—Emptying Pipes in Frosty Weather—Salt Water—Exhaust Silencers—Brick Pit, How to Construct—Exhaust-Gas Deodorizer,

PAGE

How to Connect—Connecting Circulating Water to Exhaust-pipe—Self-starters, Why Necessary —Utilizing Waste Heat of Exhaust Gases and of Cooling Water, Different Methods—Exhaust Temperature, ... 96-110

CHAPTER V.

OIL ENGINES DRIVING DYNAMOS.

Isolated Plants—Advantages of Oil Engines as Compared with Gas and Steam Engines—Installation of Plant—Foundation, How to Build, Ingredients —Correct Location of Engine and Dynamo—Belts—Balance-wheel on Armature Shaft—Power Required for Incandescent and Arc Lamps—Losses of Power by Belt and Otherwise—Regulation of Engine Required for Electric Lighting—Direct-connected Plants, Advantages of Same—Variations in Incandescent Lights, Causes, How to Remedy—Silencing Air-suction, ... 111-122

CHAPTER VI.

OIL ENGINES CONNECTED TO AIR-COMPRESSORS, WATER-PUMPS, ETC.

Direct-connected and Geared Air-compressing Outfits, with Dimensions and Pressures Obtained—Calculations of Horse-power Required—Tables of Pressures and Other Data—Efficiencies at Different Altitudes—Pumping Outfits Described in Detail, with Dimensions—How to Calculate Horse-power Required—Oil Engines Driving Ice and Refrigerating Machines, Calculations of Power Required —Friction-clutches, ... 123-138

CHAPTER VII.

INSTRUCTIONS FOR RUNNING OIL ENGINES.

PAGE

General Instructions and Remarks—Cylinder Lubricating Oil—Instructions in Detail as to Running Hornsby-Akroyd Type, the Crossley Type, the Campbell Type, and the Priestman Type of Oil Engine—General Remarks—Regulation of Speed—How to Reverse Direction of Running of Engine, with Diagrams of Valve Settings, ... 139-156

CHAPTER VIII.

REPAIRS.

Drawing Piston—Taking Off Piston-ring—Grinding in of Valves—Adjustment of Crank-shaft and Connecting-rod Bearings—How to Fit New Piston-ring to Cylinder—Fitting New Skew and Spur Gear—Renewing Governor Parts, ... 157-160

CHAPTER IX.

OIL ENGINE TROUBLES.

Ignition—Electrical Connections—Tube Igniter—Automatic Igniter—Oil Supply—Air Supply—Knocking—Loss of Power—Piston Blowing—Explosions in Silencer—Water Leakage, ... 161-167

CHAPTER X.

VARIOUS ENGINES DESCRIBED.

PAGE

General Description, with Illustrations of Different American and English Oil Engines—Method of Working—Sectional Cuts—The Crossley—The Cundall—The Campbell—The Priestman—The Mietz & Weiss—The Hornsby-Akroyd—The Diesel—The Rites Governor—Britannia Co.'s Engine—International Power Co.—The Barker, 168-199

CHAPTER XI.

PORTABLE ENGINES.

General Description of Portable Oil Engines—Portable Electric Lighting—Water Cooling Apparatus—Crossley—Mietz & Weiss—Portable Air Compressor—Hornsby-Akroyd Traction Engine, 200-205

CHAPTER XII.

LARGE SIZED ENGINES.

Comparison of Cycles—Relative Cost of Installation and Operation of Steam, Gas, and Oil Engines—Mietz & Weiss—Diesel—Hornsby-Akroyd—Sectional Views—Tests—Use of Crude Oil—Moore & Co. Vaporizer or Producer—Fairbanks Morse Engine, ... 206-229

CHAPTER XIII.

FUELS.

Description of Various Fuels—Beaumont Crude Oil—Russian and American Crude Oil—Analyses—Various Tables—California Crude-Fuel Oil, .. 230-240

CHAPTER XIV.

MISCELLANEOUS.

PAGE

Comparison of U. S. and American Measures and Weights—Various Tables—Fire Insurance—Tests of Various Engines, ... 241-251

TABLES. PART II

		PAGE
I.	Sizes of Crank-shafts, ...	27
II.	Various Air Pressures, ...	126-127
III.	Efficiencies of Air Compressors at Different Altitudes, ...	129
IV.	Mean Pressure of Diagram of Gas (Ammonia) Compressor, ...	135
V.	Tests of Priestman Oil Engine, ...	178
VI.	Tests of 25 B. H. P. Hornsby-Akroyd Oil Engine, ...	186
VII.	Relative Cost of Installation and Operation, Gas, Steam and Oil Engines, ...	209
VIII.	Tests of Diesel Engine, ...	220
IX.	Characteristics of Oils, ...	234
X.	Beaumont Oil, ...	234
XI. XII. XIII.	Characteristics of Different Oils, ...	235
XIV.	Calorific Power of Various Descriptions of Petroleum, ...	236
XV.	Composition, Physical Properties, etc., of Various Descriptions of Petroleum, ..	237
XVI.	Oil Fuel, ...	238
XVII.	Calorific Power of Crude Petroleum, ..	238
XVIII. XIX.	Tests of Various Oil Engines	248-251

PART II.

LIST OF ILLUSTRATIONS

	PAGE
Abel Oil-tester	91
Air-compressing Outfit, Portable	204
American Oil Engine Co.'s Engine	195, 196
Apparatus for Open Fire Test	91
Automatic Air Inlet-Valve	41
Barker Engine	197, 198
Beau de Rochas Cycle, Diagram	16
Britannia Co.'s Engine, Sectional View	192b
Campbell Diagrams	174
Campbell Vaporizer	10b
Campbell Type Engine	173
Cams, Air and Exhaust	36b
Connecting-rods	32a
Connecting-rod Bearings	159
Connecting-rod, Phosphor-bronze	31
Cooling Water Tower	98a
Cooling Water Tower and Radiator	98b
Cooling Radiator with Electrically Operated Fan Attachment	98b
Crank-shaft bearing	38b
Crank-shafts, Balanced	26b
Crank-shafts, Slab Type	27
Crosby Indicator	68
Crossley Diagrams	170
Crossley Vaporizer	2b, 4a
Crossley Type Engine	169
Crossley New Type Engine	170a
Cundall Type Engine	171

	PAGE
Cylinders	24a
Cylinders	26a
De la Vergne Engine, Sectional Views	226a, 226b
De la Vergne Vertical Type	183
De la Vergne Vertical Type and Air Compressor	128
De la Vergne Indicator Diagram	186
De la Vergne Indicator Diagram	228, 229
Diagram of H. P. for Air Compressing	130
Diagram of Valve-settings	60
Diagrams, Reversing Engine and Cams	155
Diesel Motor	213, 214a, 216, 219
Diesel Motor, Indicator Diagram	218
Direct-connected Air-compressing Plant (Sectional View)	124
Dynamo Fly-wheel	116
Electric Spark Igniters	4b
Engine and Dynamo, Belt-driven	112
Engine and Refrigerating Machine	132
Engine Connected to Water-pump	128b
Engine Connected to Water-pump, Small Type	131
Engine Foundation	114
Exhaust Silencing Pit	101
Exhaust Washing Device	102
Fly-wheels	36a
Foundation and Oil Tank	114a
Friction-clutch	138
Geared Air-Compressing Plant	124b
Governors	48a
Governors, Centrifugal Type	42b
Governor, Hit-and-miss Type	47
Heating Lamp	142
Heating Arrangement	109
Hill Self-recording Speed Counter	85
Hornsby-Akroyd Engine and Dynamo	118, 187, 188
Hornsby-Akroyd Horizontal Type	182a
Hornsby-Akroyd, Sectional View	210b

	PAGE
Hornsby-Akroyd Sprayer	10a
Hornsby-Akroyd, Sectional View	182b
Hornsby-Akroyd Vaporizer	2a
Hornsby-Akroyd Vertical Type	187
Hornsby-Akroyd 125 H. P.	210a
Hornsby-Akroyd 250 H. P. Oil Engine Direct Connected to Compressor	124a
Indicator Cock	66
Indicator Cards, Various	24b
Indicator Diagram	76
Indicator Diagram	77
Indicator Diagram	79
Indicator Diagram	80
Indicator Diagram	82
Indicator Diagram, Light Spring	89
Indicator Diagram, Varying Pressures	46
Indicator Diagrams, Hornsby-Akroyd	184
Indicator, Reducing Motion	67
Johnston Oil Engine	192a
Johnston Oil Engine Diagram	191
Lucke & Verplank Vaporizer	14b
Lubricator	58
Mietz & Weiss Oil Engine and Air Compressor	128a
Mietz & Weiss Oil Consumption Diagram	179
Mietz & Weiss Indicator Diagram	181
Mietz & Weiss Engine and Dynamo, Direct connected	120
Mietz & Weiss Type Engine	178a, 178b
Oil Engine with Testing Apparatus Applied	62
Oil-filter	49
Oil-pump	144
Oil Pumping Station	130a
Oil-Supply Pumps	48b
Pistons, Section of	32b
Piston with Piston-rings	56
Planimeters	72
Planimeter in position	74

	PAGE
Portable Electric Lighting Outfit	202a
Portable Oil Engine	202, 203
Priestman Engine	176
Priestman Indicator Diagrams	177
Priestman Sprayer	14
Priestman Vaporizer	13
Rites Governor	189
Self-starter	106
Silencing Device	104
Sprayers, Oil	14a, 14b
Spur-gearing	44
Starting Cam	143
Tachometer	84
Tachometer, portable	85
Testing Apparatus	64a, 64b
Testing Oil-pump	147
Traction Engine	202b
Two-cycle Plan	17
Valve-box	38a
Valve-closing Springs	42a
Valve-levers	146
Valve Mechanism	44
Valves, Air and Exhaust	42
Vaporizer, C. C. Moore & Co.	222, 224a
Vaporizer, Fairbanks-Morse	224b
Viscosometer	94
Water-circulating Pump	102
Water-cooling Tank and Connections	97
Worm Gear	43

CHAPTER I.

INTRODUCTORY—VAPORIZERS, SPRAYERS, IGNITORS, CYCLES, ETC.

THE oil engines treated of herein are internal combustion engines burning kerosene, fuel oil or crude oil, petroleum, coal oil, distillate, paraffine, etc. Such fuels have a specific gravity varying from 78° to 96° or 50° Beaume to 14° Beaume and have a flashpoint from 75° to 300° Fahr. The oil engines described are chiefly self-contained, that is, they are gas engines with the addition of a vaporizing apparatus which can convert the fuels above referred to, either in the crude state as it issues from the ground, or in a semi-refined or refined state into vapor or gas within either the vaporizers or cylinders, ignite it with the consequent evolution of the heat stored in the fuel and convert same into power.

The use of heavy oil for producing power in internal combustion engines appears to have received the attention of inventors as early as 1790, though no satisfactory practical kerosene or crude-oil engine is recorded as having been made until about 1870. Those engines using the lighter grade fuels, such as benzine, gasoline, or naphtha, were commonly used previous to the invention of the kerosene-oil engine. The prob-

lem of efficiently producing a vapor and suitable explosive mixture of air with such vapor, from these light oils was comparatively a simple matter.

With the engine required to consume crude oil or the other fuels above named having a higher boiling point than gasoline and requiring different treatment to ensure proper vaporization and to consume all parts of the heavier fuels, the problem of developing an apparatus to operate satisfactorily under all conditions and under changing loads was more complex.

The following descriptions will show how efficiently and satisfactorily the present engines operate.

IGNITERS.—The first oil engines built had their charge of vaporized oil and air ignited by means of the flame igniter, which has, however, now entirely given place to the four following means of ignition:

(*a*) Hot surface ignition, aided by compression.
(*b*) Hot tube.
(*c*) Electric igniter.
(*d*) High compression only.

The first-named type of igniter is illustrated in Fig. 1. In this instance the heated walls of the vaporizer act as the igniter, aided by the heat generated during compression of the gases. The chamber being first heated, afterward the proper temperature is maintained by the heat caused by the internal combustion of the gases. The best-known vaporizer and igniter of this type is that in the Hornsby-Akroyd Oil Engine. Various other somewhat similar devices in which sufficient heat is maintained to cause ignition automatically are also now being made.

The second type, that of the hot tube, is shown in

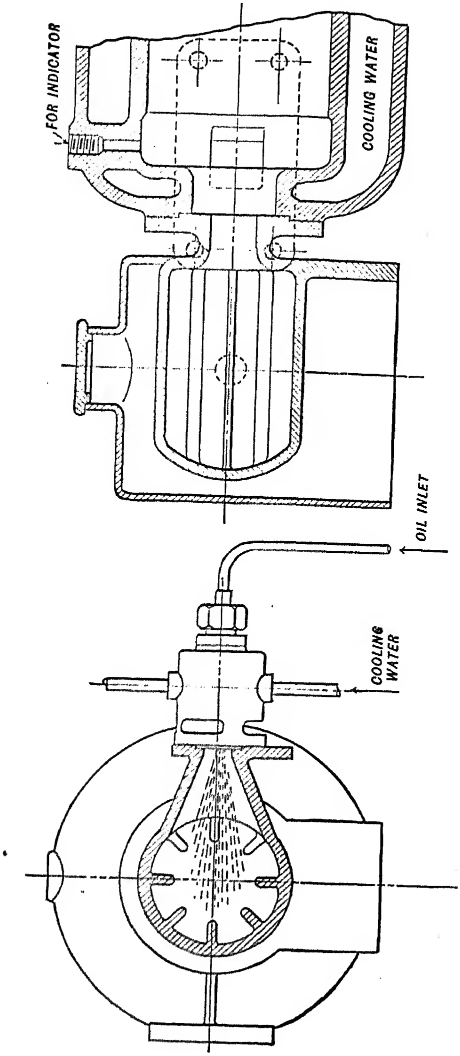
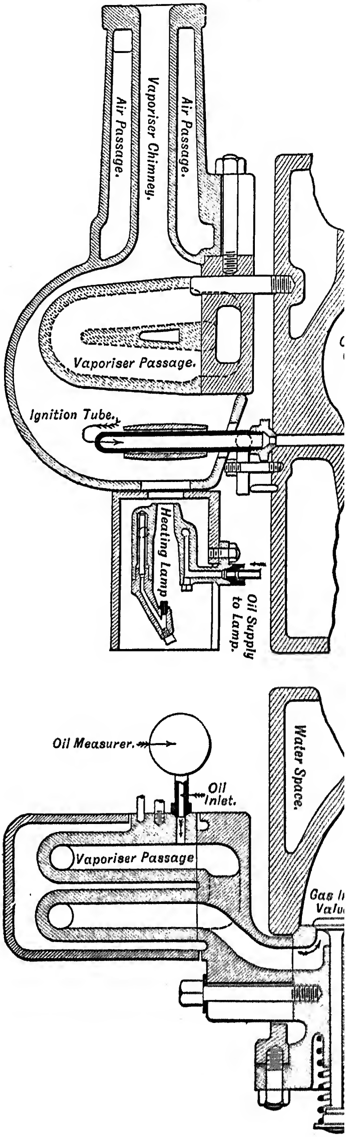


FIG. 1.


Air Passage.
Vaporiser Chimney.
Air Passage.
Vaporiser Passage.
Ignition Tube.
Heating Lamp
Oil Supply to Lamp.
Oil Measurer.
Oil Inlet.
Water Space.
Vaporiser Passage

Figs. 2 and 3. This igniter consists simply of a porcelain or metal tube fitted into the vaporizer or cylinder wall. It is closed at one end, the other end being open to the cylinder. It is heated by a lamp, as shown in Figs. 2 and 3, over part of its length. When compression due to the inward stroke of the piston takes place in the cylinder the explosive mixture is compressed into the tube and is ignited by coming in contact with the heated portion of it. Porcelain or nickel-steel tubes are preferable to wrought iron, all of which substances are used for this purpose.

The electric igniter, which is at present more largely used for gas and gasoline engines than for oil engines, is shown in Fig. 4. Those illustrated are known as the "jump-spark" and the make-and-break types.

The jump-spark (Fig. 4) is preferred for high speeds, as it has no moving parts inside the cylinder. With this type the igniter plug containing the terminals is screwed into the cylinder cover. The method of making electrical connections is shown in principle at Fig. 4. Connection is made from the battery through the primary circuit of the Rhumkorff or spark coil to the completely insulated spring which is operated by the cam. The other connection passes from the battery to the other spring operated by the cam-shaft or other moving part of the engine. The electrodes or terminals of the plug are connected to the secondary circuit. In operation where a vibrator is used in connection with the spark coil the cam at the proper time of sparking closes the circuit, causing a series of sparks to jump across the terminals in the cylinder and ignite the gases.

The make-and-break type of igniter is shown in Fig. 4*a*. This type consists of one well-insulated stationary terminal and one terminal *H* mechanically operated. The ignition is caused by the separation of the two terminals, which produces a spark between them. Fig. 4*a* shows this igniter in connection with a magneto oscillator, which is frequently employed to furnish electrical current instead of the battery. With this apparatus the current is generated by the quick movement of the inductor, which takes the place of the armature in the ordinary dynamo, and which is caused to partly revolve by movement of the arm suitably actuated from the cam-shaft or other moving part of the engine. The magneto is a very simple device, consisting only of stationary steel magnets *K*, a cast-iron inductor which takes the place of the ordinary armature, and two coils imbedded in the frame. The action is as follows: The inductor arm *C* is raised by the roller *A* on the disc *B* attached to cam-shaft. The spring *D*, shown in Fig. 4*a*, is compressed. When the arm is released the inductor has a quick, oscillating motion, caused by spring *D*, which produces a strong electrical current. This current passes through connection *J* to insulated igniter point, and through the movable electrode *G* back to the induction apparatus. The movement of inductor lever by the heavy spring allows the collar on rod *E* to hit the arm attached to movable electrode, thus separating the two electrodes and causing a spark to pass between them.

A spark plug is shown in section at Fig. 4*b*, made by A. W. King. Advantages are claimed for this type

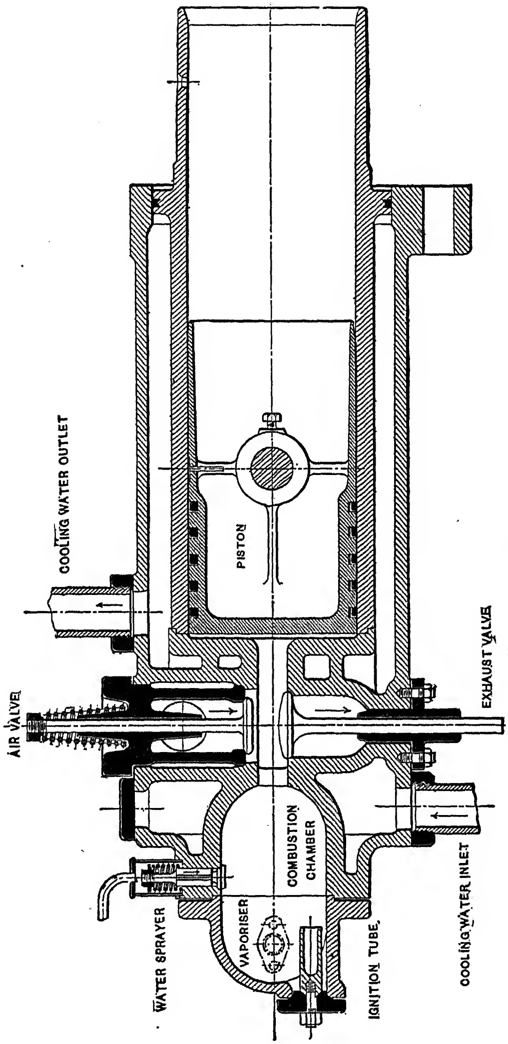

COOLING WATER OUTLET
PISTON
AIR VALVE
EXHAUST VALVE
COMBUSTION CHAMBER
WATER SPRAYER
VAPORISER
IGNITION TUBE
COOLING WATER INLET

FIG. 3.

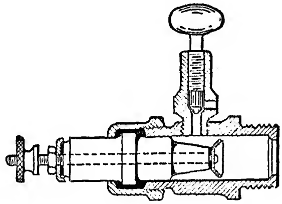


FIG. 4*b*.

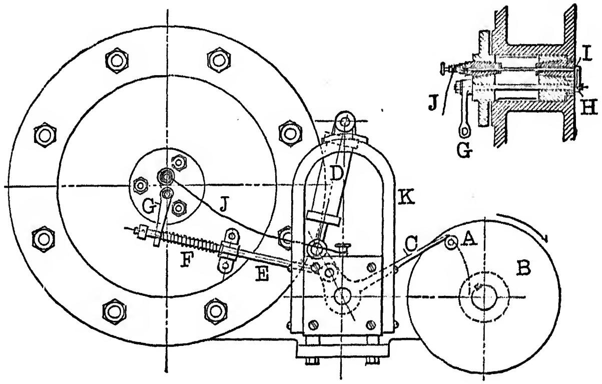


FIG. 4*a*.

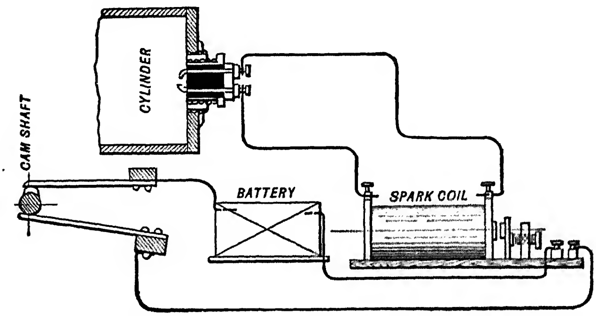


FIG. 4.

of plug because of the increased sparking surface of the terminal, which is formed of an inner knife-edged disc placed concentric within a thick-wall chamber, which constitutes the outer terminal. Other forms of electrical igniters are the New Standard and the Splitdorf jump-spark apparatus.

The fourth-named type of ignition, that due to compression in the cylinder alone, is found only with the Diesel motor.

Advantages are claimed for each of these igniting devices by the various manufacturers using them. The electrical igniter is easily controlled and is reliable, but the batteries in unskilled hands sometimes give trouble, and it is essential that the parts forming the contacts be kept clean and in good condition.

The tube igniter always requires heating by the external heating lamp, upon which it is dependent, like all types of vaporizers which require external heat; so likewise is also the tube dependent entirely upon it. The former difficulty with ignition tubes and their frequent bursting has now been minimized by the use of nickel alloy, porcelain or other material more suitable than wrought iron for this purpose.

The hot surface type of igniter formerly gave trouble caused by its temperature cooling down at light loads. This type, however, which has now been adopted in various forms, has been designed to overcome this difficulty, and can now be relied upon to keep hot when running at light loads.

VAPORIZERS.—As already stated, the problem of efficiently vaporizing petroleum was the most difficult feature to encounter in designing oil engines.

The present universal use of heavy oil engines is complete evidence of how any former difficulty has been thoroughly overcome, and examination of the various modern vaporizers shows extreme simplicity in operation.

The fuels used in the oil engines here discussed (crude oil, kerosene, etc.), in order to be properly vaporized, require to be broken up into the form of mist or oil vapor by spraying, or by a current of air, and then heated to a temperature above the boiling point. The oil vapor must then be thoroughly mixed with air, in order to procure complete combustion. This process is performed by various methods, as is shown in the following description of vaporizers.

The composition of various fuels is discussed in Chapter XIII.

Several oil engines having a method of vaporization are now made where the oil is injected directly into the cylinder or where it is inhaled with the air, and where both are closely regulated similar to the Priestman type of oil engine. The mixture of oil vapor and air being carried on by compression in the cylinder, ignition is caused by an electric or tube igniter. The heat from the exhaust is utilized to raise the temperature of the chamber through which the oil passes to the cylinder, which, with the heat caused by compression, is sufficient to cause vaporization and a proper mixing with the air to form an explosive mixture, the chamber, which is heated by the exhaust in operation being first heated by a lamp.

Theoretically, the amount of air required for each

pound of kerosene or oil vapor is approximately 200 cubic feet at 60° Fahr. atmospheric pressure. From calculation of the amount of air taken into the cylinder, it will, however, be noted that this amount in practice is much greater. In some instances it is more than twice that amount, or 400 cubic feet. This greater volume of air is required owing to the presence in the cylinder, in operation, of a residue of the burnt products of previous explosions and to other impurities causing the efficient combustion of the oxygen of the air with the oil vapor to be somewhat retarded.

A method of starting the oil engine has of recent years been used in which alcohol, gasoline, or naphtha is burnt for a few minutes instead of kerosene. This method is advantageous in that the engine when cold can be started without the use of external heater. The lighter fuel is supplied to the vaporizer or cylinder until the vaporizing attachment has become heated by internal combustion to the temperature necessary for vaporizing the heavier fuel; then the fuel supply is changed, the supply of lighter fuel being stopped. Where an automatic igniter or vaporizer of Type 4 is used an independent electric igniter is employed to ignite the gases, and which is only in action until the vaporizer is heated.

The different types of vaporizers have been classified as follows:

1. The vaporizer into which the charge of oil is injected by a spraying nozzle being connected to cylinder through a valve.

2. That into which the oil is injected, together with some air, the larger volume of air, however, entering the cylinder through separate valve.

3. That vaporizer in which the oil and all the air supply (passing over it) is injected, but being without spraying device.

4. The type into which oil is injected directly, air being drawn into the cylinder by means of a separate valve, the explosive mixture being formed only with compression.

With each type of vaporizer some advantage is claimed, but corresponding disadvantage can perhaps be named. For instance, in type 1, though the mixture of oil and air is more complete, and the vaporizing probably greater than in the other types, yet the system of having an explosive mixture at any other place than in the cylinder and at any other period than at the time of actual ignition may be urged as a great disadvantage to this system.

With class 4 the mixture of air and oil may not be so complete, and the initial pressure in the cylinder consequent upon explosion less than the pressure obtained with other types; yet the extreme simplicity of this type is an advantage in daily use which cannot be overestimated.

With class 2 the highest mean effective pressure is obtained and the lowest consumption of oil per H. P. is recorded, but where a heating lamp burning continuously is required then on the heating lamp depends the efficiency of the engine itself.

LUCKE AND VERPLANK VAPORIZER.—An apparatus for vaporizing crude or fuel oil is shown at Fig. *7c*; it consists of a chamber containing liquid fuel surrounded

by an exhaust heating jacket. The fuel is maintained at a temperature corresponding to its boiling point, and freely gives up vapor without overheating or carbonizing. The piping arrangement allows liquid oil to be constantly present in the chamber. The fuel enters at the bottom, and after vaporization, some is blown off through the connection leading to the condenser while the rest enters a mixing and proportioning valve supplying the engine with correct clean explosive mixture. If the load on the engine does not require the full amount of vapor, it is condensed. The lower blow-off cock allows the liquid residue carbon to be disposed of when crude or fuel oils are used. When using distillate, kerosene, etc., the blow-off is dispensed with. Fig. 7*c* shows the pressure type of vaporizer, but by breaking the pipe between condenser and feed and inserting a constant level open cap, vapor is generated at atmospheric pressure, then one or both check valves are omitted.

The Hornsby-Akroyd vaporizer is shown at Fig. 1, and also as it is at present manufactured in Fig. 76, which illustrates a complete section of this engine. The oil in this method of vaporizing is injected through the spray nipple, as shown in Fig. 5, directly into the vaporizer by the oil-supply pump. The injection of oil into the vaporizer takes place only during the air-suction stroke. The lever which actuates the air-valve also simultaneously operates the oil-pump. When the piston is at the outward end of the cylinder, the suction period being then completed, the cylinder is filled with atmospheric air, and the vaporizing chamber, which is at all times open to the cylinder, is also at the same time filled with oil vapor.

The compression stroke of the piston then com-

mences; the atmospheric air in the cylinder is thus driven through the contracted opening between the cylinder and the vaporizer into the vaporizer itself, already filled with the oil vapor. The oil enters the vaporizer in the form of a thin spray or sprays and impinges on the cast-iron vaporizer wall on the opposite side, and then forms a vapor which afterwards mixes with air. Two forms of oil injectors are shown in the accompanying illustration, Fig. 5*a* being that used in connection with the later type of Hornsby-Akroyd vaporizer, which is partly water-jacketed; in this type a circular passage is made through the water-jacketed part of the vaporizer, into which the oil-spray sleeve is fitted. The water circulating around the vaporizer maintains the whole at a low temperature. Fig. 5 shows the older type of oil inlet sleeve and sprayer. Another form of oil injector made by the English makers of this engine is shown at Fig. 95. In this type the water jacket is eliminated, the heat being carried away by the surrounding air and by the fuel passing through it as it is pumped to the vaporizer. The steel spray nozzle in this type is a loose piece, being held in place by the pressure of the studs holding the sleeve containing the valve against the vaporizer. After the oil is injected into the vaporizer the compression stroke commences as this proceeds; the mixture, which at first is too rich to explode in the vaporizer, gradually becomes more diluted with the air, and when the compression stroke is completed the mixture of oil, vapor and air attains proper explosive proportions. The mixture is then ignited simply by the hot walls of this

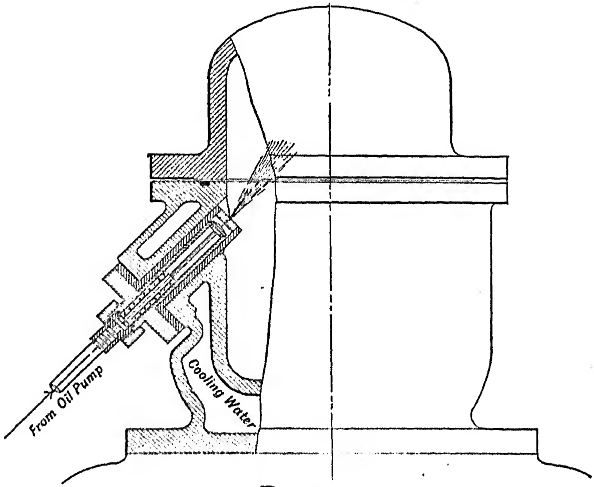
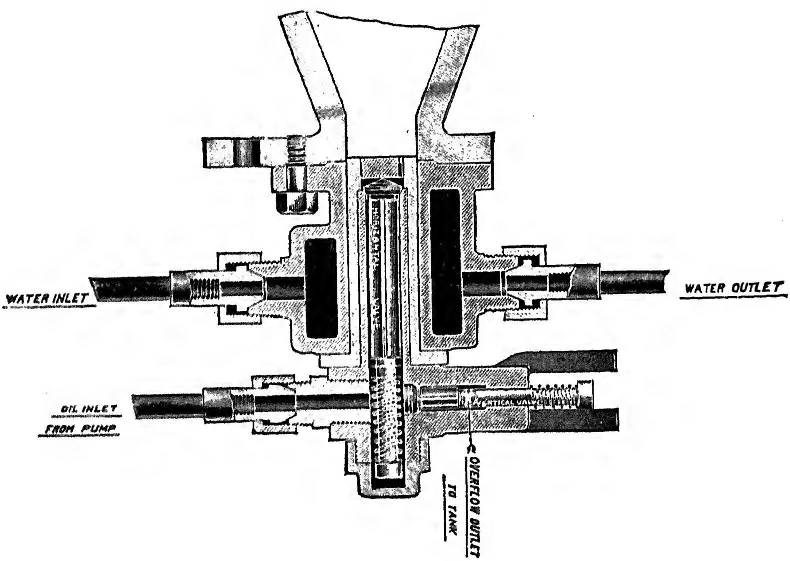

From Oil Pump
Cooling Water

FIG 5*a*.


WATER INLET
WATER OUTLET
OIL INLET
FROM PUMP
OVERFLOW OUTLET
TO TANK

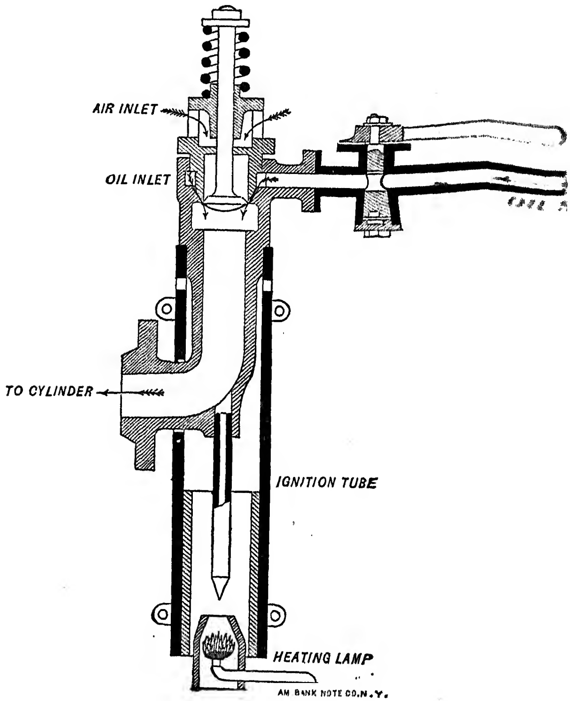

AIR INLET
OIL INLET
TO CYLINDER
IGNITION TUBE
HEATING LAMP
AM BANK NOTE CO.N.Y.

FIG. 6.

same vaporizing chamber and also by the heat generated by compression. No other means of ignition is necessary. No heating lamp is required to maintain the necessary temperature of this vaporizer; a lamp is, however, required to heat it for a few minutes before starting.

THE CROSSLEY method of vaporizing. This vaporizer is shown in section in Fig. 2. It consists of three main parts, the body, the passages, and the chimney cover. There are no valves about the vaporizer itself; it is arranged to keep hot, and while not in contact with the cooled cylinder is near to the vapor inlet valve to which it delivers its charges. The passages inside which vaporization of the oil takes place are detachable.

The wrought-iron ignition tube is placed below the vaporizer communicating directly with the cylinder. A heating lamp is always required to heat the vaporizer and maintain the ignition tube at proper red heat. The method of vaporizing is as follows:

When the suction stroke of the piston commences the oil inlet valve is automatically lifted from its seat and allows oil to be drawn into the vaporizer through it. The vaporizer blocks having been heated by the independent lamp, and likewise the chimney being hot also, heated air is drawn in passing first through the apertures in the sides of the chimney communicating with the passages of vaporizer blocks. The air is thus thoroughly heated, and next it passes over the heated cast-iron blocks. To these blocks the oil also flows from the oil measurer. The heated air here mingles with

the oil and vaporizes it, and the two together properly mixed are drawn into the cylinder through the vapor valve. Simultaneously, while the above process of vaporization is proceeding, air is also entering the cylinder through the air-inlet valve on the top of the cylinder. Thus, when the suction stroke of the piston is completed the cylinder is full of heated oil vapor drawn in through the vapor valve, too rich to explode by itself, and also atmospheric air drawn in through the air valve. Both elements are then compressed by the inward stroke of the piston completing the mixture of the oil, vapor and air.

Fig. 3 shows the latest type of Crossley vaporizer which only requires heating when starting the engine. The fuel is injected directly into the vaporizer through the sprayer shown at *C*, Fig. *7a*, placed on the side of the vaporizer. A small amount of water with some air also enters this vaporizer.

Fig. 6 represents the Campbell vaporizer in section. The fuel oil is fed to the vaporizer by gravitation from the fuel tank placed above the engine-cylinder, and enters the vaporizer with the incoming air. At the beginning of the suction stroke the automatic air-inlet valve is opened by the partial vacuum in the cylinder, and the oil which has entered through the small holes at the inlet valve is drawn through the heated vaporizer into the cylinder. At the compression stroke the mixture of the vapor is completed, and being forced into the ignition tube is ignited in the ordinary way. The ignition tube is heated by heating lamp fed by gravitation from the oil tank. The same lamp also heats the

vaporizer as well as the tube. The governing is effected by allowing the exhaust-valve to remain open when the normal speed is exceeded; consequently no charge is in that case drawn into the cylinder.

SPRAYERS.—The oil-spraying device of an oil engine is an important feature. In some engines the fuel is sprayed alone into the vaporizer. In others with the highest thermal efficiency compressed air is injected with the fuel. Various sprayers are shown at Fig. 7*a* and 7*b*. That at *A* is positively operated and allows air and fuel to enter the vaporizer together; those at *B* and *C* are automatic and only fuel is sprayed.

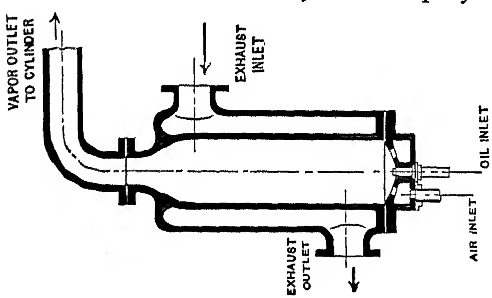


FIG. 6*a*

The method of vaporizing the oil with the PRIESTMAN engine is as follows:

The oil is stored under pressure in the fuel-tank, which pressure is created by the separate air-pump actuated from the cam-shaft. The oil is thus forced to the sprayer, which device is shown in Fig. 6*a*, where it meets a further supply of air. The mixing of the air and oil takes place just as both elements are injected

into the vaporizing chamber, as shown in Fig. 6*a*. The heating of the vaporizer is first accomplished with separate lamp; afterward, when the engine is working, the exhaust gases heat the vaporizer by being carried around in the outside passage of the vaporizer cham-

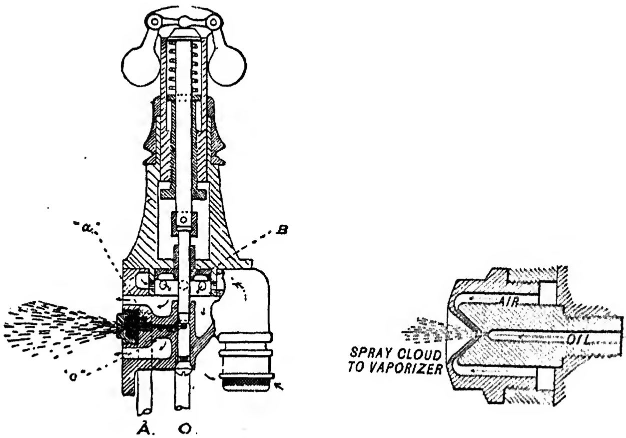
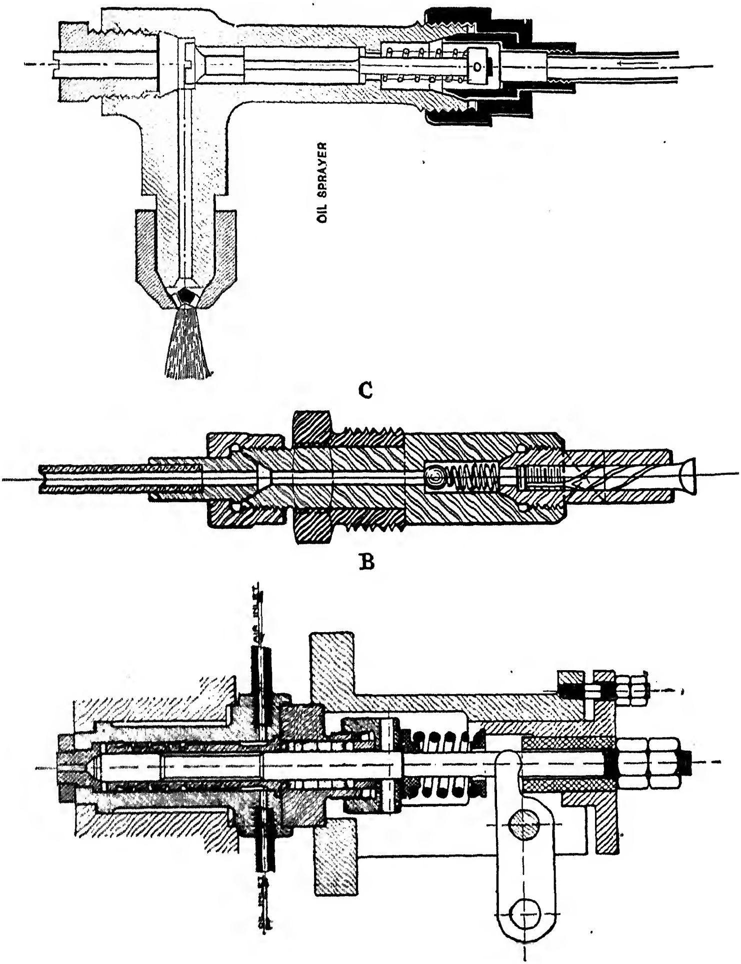


FIG. 7.

"*A*"—Air pump connection. "*a*"—Air passage to spraymaker. "*O*"—Oil tank connection. "*o*"—Oil passage to spraymaker. "*B*"—Supplementary air valve.

ber, as shown in Fig. 6*a*. On the outward or suction stroke of the piston the mixture of oil vapor and air already formed and heated in the vaporizer is drawn into the cylinder through the automatic inlet-valve shown on the left of Fig. 6*a*. The compression stroke



A

FIG. 7*a*.

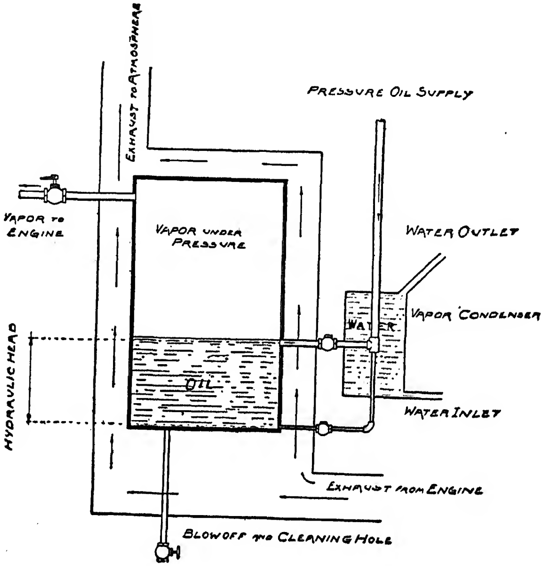

EXHAUST TO ATMOSPHERE
PRESSURE OIL SUPPLY
VAPOR TO ENGINE
VAPOR UNDER PRESSURE
WATER OUTLET
WATER
VAPOR CONDENSER
HYDRAULIC HEAD
OIL
WATER INLET
EXHAUST FROM ENGINE
BLOWOFF AND CLEANING HOLE

FIG. 7c.

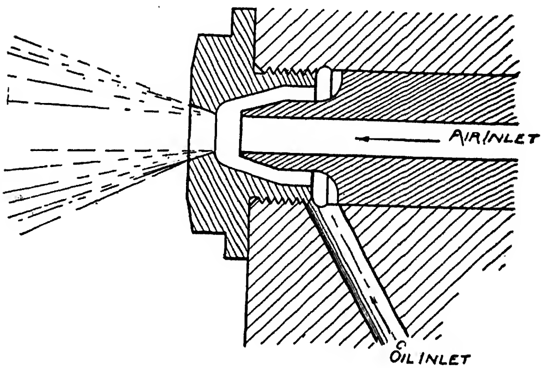

AIR INLET
OIL INLET

FIG. 7b.

then takes place in the ordinary course of the Beau de Rochas cycle.

The governing is effected by means of the pendulum or centrifugal governor, shown at Fig. 7, controlling the amount of air entering the vaporizer as well as reducing the supply of oil simultaneously. Thus, the explosive mixture is always composed of the same proportions of air and oil, but as the supply of air is thus curtailed the compression in the cylinder is also necessarily reduced when the engine is working at half or light load. The governor thus varies the pressure of the explosion, reducing it when necessary, but not causing at any time the complete omission of an explosion.

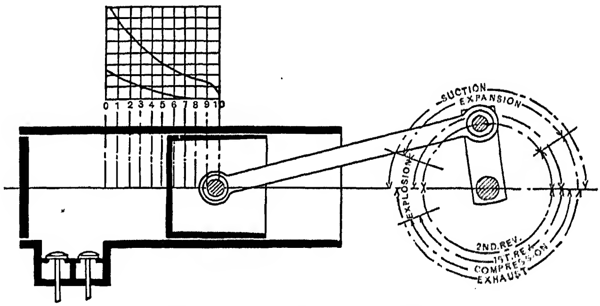
The system of throttling the pressure, somewhat similar to a steam engine, produces very steady running.

By this system a thorough vaporization of the oil takes place.

The ignition of the gases is caused by electric spark-igniter, the spark being timed by contact-pieces actuated from the cam-shaft and horizontal rod actuating the exhaust-valve, and is of the "jump-spark" type as shown in Fig. 4.

The oil engines now in use and herein described are designed with their valve mechanisms arranged to work either on the Beau de Rochas cycle, or on the two-cycle system. These two cycles are variously designated, the former being generally known as the Otto cycle, the four-cycle, and sometimes, but erroneously, the two-cycle. Correctly, it should be named the Beau

de Rochas cycle after its inventor. The other cycle is generally known as the "two-cycle," or sometimes as the "single cycle," the first designation, however, being correct. With those engines working on the Beau de Rochas cycle, which includes now many if not all the leading and best known types of engine,



THE BEAU DE ROCHAS CYCLE.

the cycle of operation of the valves is as follows:

(*a*) Drawing in the air and fuel during the first outward stroke of the piston at atmospheric pressure.

(*b*) Compression of the mixture during the first return stroke of the piston.

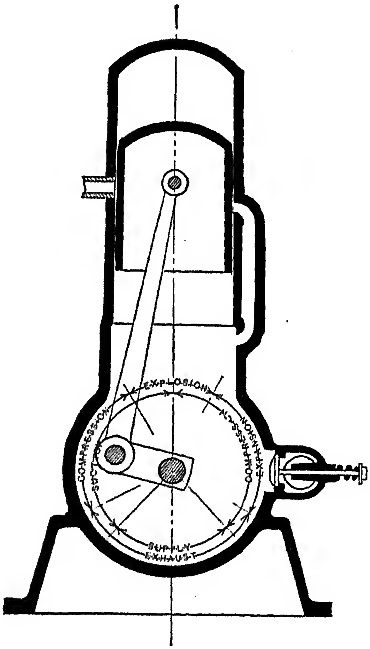
(*c*) Ignition of the charge and expansion in the cylinder during second outward stroke of the piston.

(*d*) Exhausting, the products of combustion being expelled during the second return stroke of the piston.

These operations are clearly shown in the accompanying illustration, and thus, in this system, the one cycle is completed in two revolutions of the crank-

shaft or during four strokes of the piston. The impulse at the piston is obtained only once during the two revolutions.

The second system, named "two-cycle," is com-



THE TWO-CYCLE PLAN.

pleted in one revolution, or every two strokes of the piston, and is also clearly shown by the accompanying illustration. The operation of this type is as follows:

(*a*) During the first part of the outward stroke of the piston—that is, until the piston uncovers the exhaust-port—expansion is taking place. When the exhaust-port is opened the products of combustion are expelled; the piston then moves a little farther forward and uncovers the air-inlet port communicating with the crank chamber. The air at slight pressure at once rushes into the cylinder, assisting the expulsion of the burnt gases, and filling the cylinder with air already compressed to five or six pounds in the crank chamber; this completes the first stroke of this cycle.

(*b*) The next stroke (being the inward stroke of the piston) the supply of incoming air and fuel is first taken in; then compression of the charge takes place. Ignition follows when the piston reaches the back end. These two strokes of the piston, or one revolution of the crank-shaft, completes this cycle of operation.

Advantages and Disadvantages of Both Cycles.

The Beau de Rochas cycle engine, having only one impulse during two revolutions, requires the dimension of the cylinder to be greater in order to obtain a given power than would be required with the two-cycle system. Large and heavy fly-wheels must also be fitted to the engine in order to maintain an even speed of the crank-shaft. On the other hand, this cycle has many advantages. The explosion is controlled more readily. The idle stroke of the inlet air cools the cylinder and allows sufficient time to entirely expel the products of combustion, and with this sys-

tem no outside air-pump is required, nor is there any fear of the compression being irregular by leakage in the crank chamber or otherwise.

With the two-cycle system air must in some way be independently compressed. If this is accomplished in the crank chamber, then leakage may occur and bad combustion follow, with accompanying bad results to valves and piston. More cooling water is also needed to cool the cylinder, and the proper lubrication of the piston may consequently be very difficult to accomplish. With this system steadier running is obtained, nor are the heavy fly-wheels required as with the engines of the Beau de Rochas cycle.

Large sized oil engines by all leading makers are now made of the four (or Beau de Rochas) cycle. Few if any two-cycle oil engines are now on the market where over 35 B. H. P. is developed in one cylinder. The increased volume of heated gases or vapor in the larger diameter cylinder precludes the successful operation of the two-cycle type where the explosion occurring each revolution render the cylinder difficult of proper cooling. In such engines where the pressure of compression takes place in the crank chamber, difficulty is also experienced with the heating of crank and other bearings. In the smaller sizes the two-cycle type has many advantages—notably greater frequency of impulse, decreased weight per H. P., elimination of exhaust valves and valve motion. From tables of tests* it will be noted the economy of the four-cycle is higher than that of the two-cycle type.

*See pages 249 to 252.

CHAPTER II.

DESIGN AND CONSTRUCTION OF OIL ENGINES.

THE designing of an oil engine is generally a different procedure from that of designing a gas engine. It is true, the oil engine is a gas engine in the strict sense of the term, but with the gas engine proper, the fuel enters its cylinder or mixing chamber in a gaseous state ready for mixture with the air. The power which the gas engine will develop can more readily be calculated when the clearance and pressure of compression before the explosion is known than with the oil engine.

The special apparatus which is the most important part of the oil engine is the vaporizer. The different types of vaporizers and the various methods of vaporizing the fuel have already been described and explained in Chapter I.

In practically all the oil engines herein described the vaporizing apparatus is self-contained in the engine and part of it. Before the pressures which will be developed in the cylinder can be accurately computed, experiments may be necessary to develop the allowable maximum pressure of compression which can be used to obtain properly timed ignition, complete combustion and highest fuel economy.

These remarks are particularly applicable to the type of oil engine having automatic or "hot surface" ignition. In those engines where the electric ignitor or

other mechanically controlled ignitor is used, or in the type where the injection of the fuel takes place after compression is completed, the exact timing of ignition is positively controlled and with the engine in proper working order in other respects pre-ignition cannot take place which might result with the type having automatic or "hot surface" ignition.

In this chapter it is intended only to describe as fully as possible the practical details of the construction of the oil engine. For a theoretical discussion of the thermodynamics of the internal combustion engine, the reader is referred to those works devoted to that subject.*

Briefly referred to, the ideal heat engine converts into work the fraction of heat

$$\frac{T_1 - T_2}{T_1}.$$

Where T_1 = absolute initial temperature or receptive temperature.

T_2 = absolute final temperature or rejective temperature.

The oil engine, like all other heat engines, converts into work that amount of heat being the difference between the initial temperature or heat received and the final temperature or heat equivalent of exhaust and other losses.

Thus

Heat evolved = work† + heat and other losses.

*The Theta Phi Diagram by H. A. Golding; the Steam Engine by J. H. Cotterill, and Heat Engines by Prof. Ewing.

†Heat equivalent of work is I. B. T. U. = 778 Foot pounds.

In order therefore to obtain the greatest economy, the greatest range of temperature must be allowed between the initial and final temperatures. For this reason the progress towards higher economy witnessed in recent years in the oil and gas engine has been largely if not entirely effected by the use of greater pressures of compression before ignition, where the initial temperature which is a measure of the heat received by the engine has been increased, while the final temperature has remained with little or no increase, the range between being accordingly increased.

HEAT LOSSES.—In the equation above, the heat or other losses may be classified as follows: 1. Friction in the mechanical movements of the engine itself. 2. Losses of heat through the cylinder and other water jackets. 3. Radiation. 4. Loss through exhaust gases. 5. Leakage and other losses.

INTERNAL COMBUSTION ENGINES are of substantial design in order to withstand the continual shock and vibrations incident thereto, and should pre-eminently be as accessible as possible in the working parts, which may require adjustment from time to time when in actual service. The starting gear and other parts to be handled by the attendant when starting and running should be placed in close proximity to each other.

Simplicity in construction is, in the writer's opinion, the essential feature of an oil engine. Above all other prime movers, the oil engine is a machine intended for use in any part of the world where its fuel is obtainable, and where, perhaps, no mechanic is available.

Accordingly, all the valves should be arranged so as to be easily removed for examination and repairs. The spraying and igniting device, as well as the vaporizer, should be so designed as to facilitate removal and repairs. In short, an oil engine, to be successful mechanically and commercially, should be so constructed that it can be successfully worked, cleaned and adjusted by entirely unskilled attendants.

THE INDICATED HORSE-POWER (I. H. P.) or total power developed by the engine is arrived at by the formula

$$\text{I. H. P.} = \frac{PLAN}{33,000}.$$

Where P = mean effective pressure in lbs. per sq. in.

L = length of stroke in feet.

A = effective area of piston in sq. in.

N = number of explosions per minute.

THE BRAKE OR ACTUAL HORSE-POWER (B. H. P.) developed by the engine is the I. H. P. less the friction in the engine itself and depends upon the amount of power absorbed. The mechanical efficiency of the engine (see page 86) is found by the formula

$$\text{Mech. Effi. (E)} = \frac{\text{B. H. P.}}{\text{I. H. P.}}.$$

In determining the diameter of the cylinder of an engine to furnish a required actual or Brake H. P., the diameter of the cylinder must allow for the friction losses, the mechanical efficiency being usually 80% to 85%.

The mean effective pressure (M. E. P.) may be arrived at by the following formulæ in existing engines:

$$\text{Mean effective pressure} = \frac{\text{B. H. P.} \times 396.000}{E \times V \times N}$$

$E =$ Mechanical efficiency, usually about 0.80.
$V =$ Volume piston displacement in cubic inches.
$N =$ Number of explosions per minute.

For multicylinder engines, the M. E. P. can be determined by considering the B. H. P. for one cylinder only.

The accompanying diagrams, Fig. 8*b*, are taken from different makes of oil engines which have various pressures of compression. It will be seen that while there is a certain comparison between the compression pressure and the maximum and mean effective of the oil engine the rules laid down for the gas engine do not altogether apply to the oil engine.

The formulæ given hereafter are those in many instances used for the designing of gas engines. The dimensions of the reciprocating parts are frequently, however, increased somewhat for the oil engine, especially with the type having hot surface or automatic methods of ignition.

Cylinders.—Cylinders of different types are shown at Figs. 8, 8*a*, and 9. Where the cylinder is made in two parts the inner liner is held at the back end only, the front joint being made with rubber rings. This leaves the inner sleeve free to expand lengthwise and

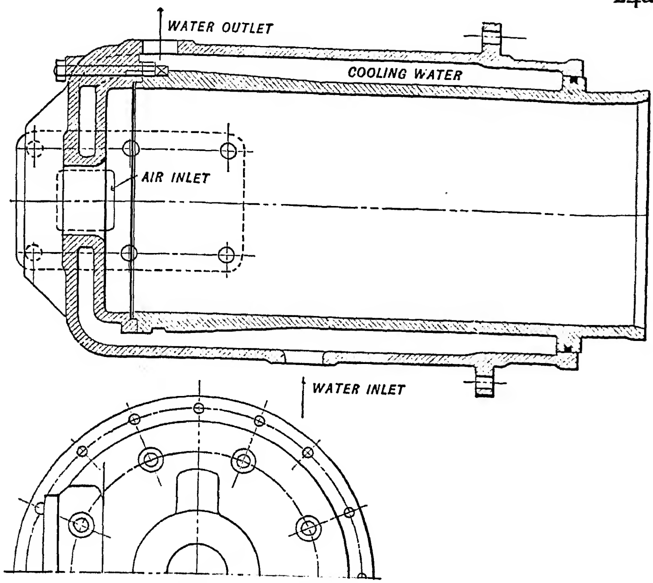


FIG. 8*a*.

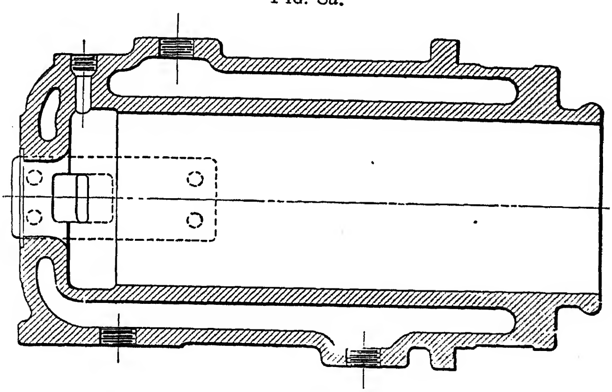


FIG. 8.

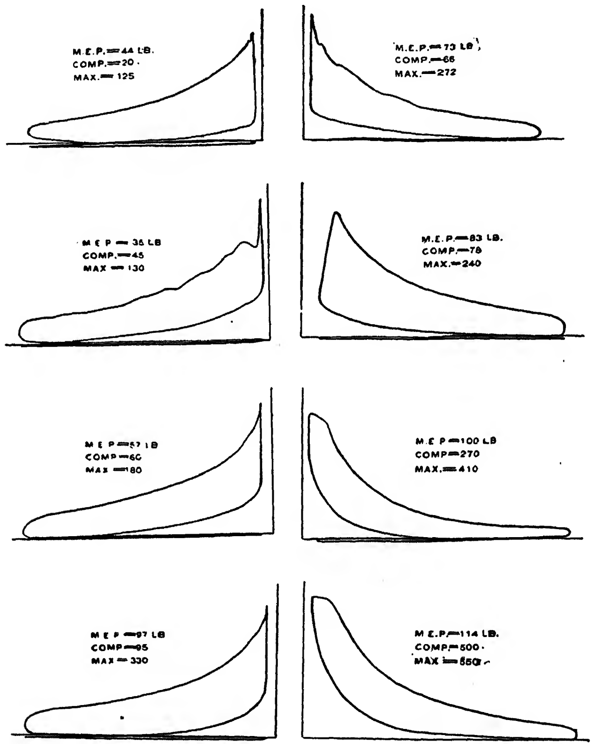

M.E.P.=44 LB.
COMP.=20.
MAX.=125
M.E.P.=73 LB.
COMP.=66
MAX.=272
M E P = 35 LB
COMP.=45
MAX = 130
M.E.P.=83 LB.
COMP.=78
MAX.=240
M E P =57 LB
COMP =60
MAX =180
M.E P =100 LB
COMP=270
MAX.=410
M E P =97 LB
COMP =95
MAX = 330
M E.P.=114 LB.
COMP.=500.
MAX =550

FIG. 8*b*.

also allows the strain of the explosion to be transmitted only through the outer cylinder. Except for the larger-sized engines of over 40 H. P., the cylinder made in one piece is very satisfactory. The circulating water space around the cylinder is made as is shown in Figs. 8 and 9, being $\frac{3}{4}''$ to $1\frac{1}{2}''$ deep, the water inlet and outer pipes being so arranged as to allow free and efficient circulation of the cooling water around the cylinder. By some manufacturers this space for water is arranged to cool only that part of the cylinder covering the travel of the piston-rings, instead of the whole cylinder, as here shown. Other cylinders are cast in one piece with the frame or bed-plate having internal sleeve. This arrangement has, among other advantages, that of cheapness, but it has the disadvantage that if the cylinder for any reason should require renewing the whole frame must be renewed with it.

The cylinder cover is made in some engines with the valves, air-inlet valve housing or guide inserted into it, and with space also in the larger-sized engines arranged for cooling water-jacket. Other engines have the igniter placed in the cover, while cylinders of the type shown in Fig. 8 require no cover, the vaporizer flange closing the contracted hole in the end of the cylinder.

Cylinder Clearance.—The percentage of clearance or clearance volume in the cylinder and combustion space may be arrived at by the following:

$$V_c = \frac{.785\, d^2 s}{(P_c)^{\frac{3}{4}} - 1}.$$

Where V_c = clearance volume in cubic inches.

P_c = compression pressure in atmospheres

$$= \frac{\text{absolute pressure}}{14.7}.$$

d = diameter cylinder in inches.

s = length of stroke in inches.

The clearance allowed with the oil engine will depend upon the type of vaporizer and the method of vaporizing adopted, on the timing of the injection of fuel, the pressure of compression and the clearance may finally have to be modified to procure the best results as shown by the indicator card.

STROKE.—The ratio of length of stroke to diameter of cylinder varies with different types of engines. The maximum speed of piston allowable is considered 900 ft. per min. In small high speed engines the

$$\frac{\text{length of stroke}}{\text{diameter of cylinder}} = 1. \text{ to } 1.3.$$

For medium sized engines this ratio is 1.3 to 1.6, while in larger engines the ratio is sometimes as large as 1.8 or 2.

THE CRANK-SHAFT of an oil engine must be made of sufficient strength not only to withstand the sudden pressure due to ordinary explosion, but also to withstand the strain consequent upon the greater explosive pressure which may possibly be caused by previous missed explosions. The crank-shaft is proportioned in relation to the area of the cylinder and the maximum pressure of explosion and the length of stroke. Oil-engine crank-shafts are usually made of the "slab type," as shown in Fig. 10. It has been said of explosive engines that their comparative efficiency may be to an extent

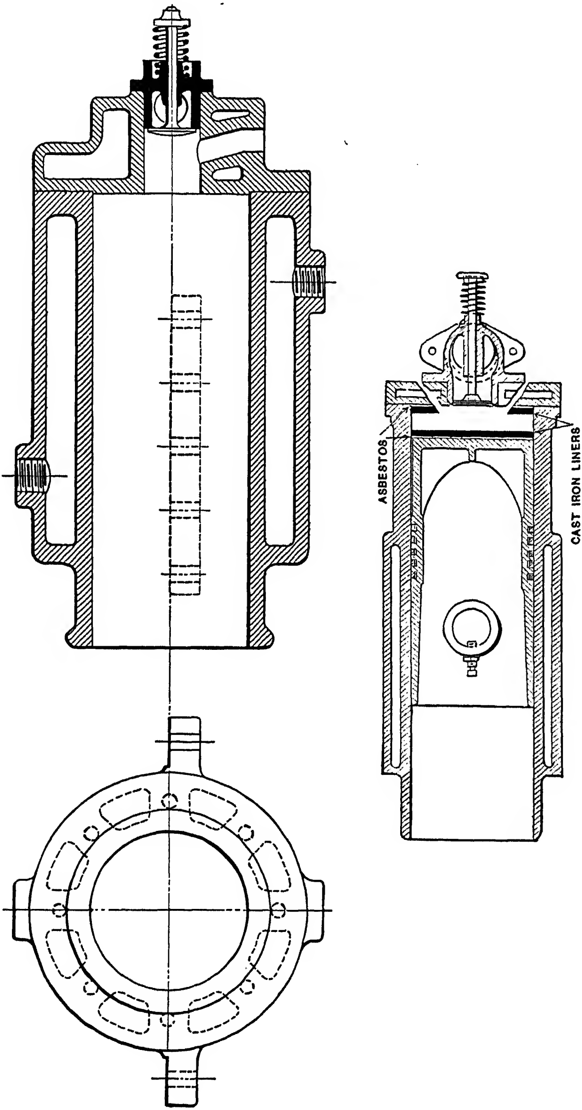


FIG. 9.

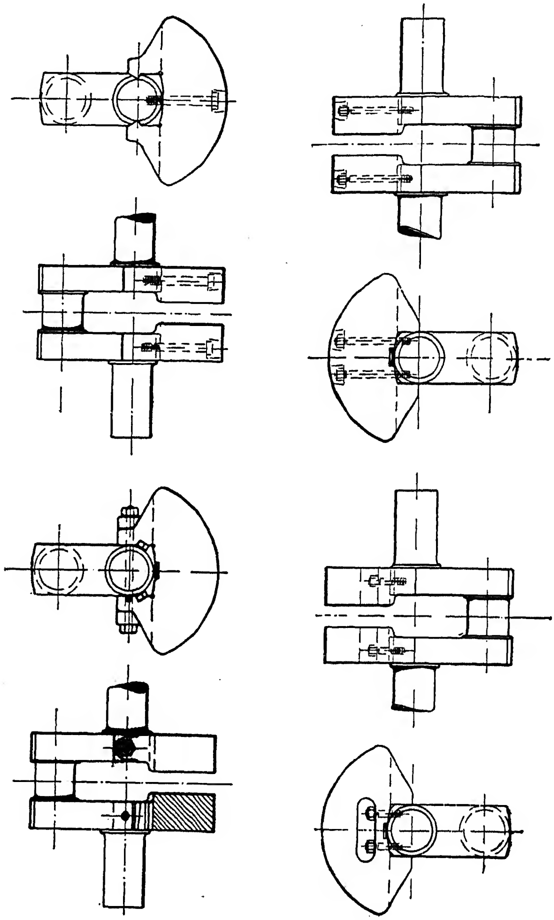


FIG. 11.

gauged by the strength of the crank-shaft, because if the crank-shaft is of too small dimensions, it will spring with each explosion, causing the fly-wheels to run out of truth and also uneven wear of the bearings. Table I. gives a list of dimensions of crank-shafts of both oil and gas engines which are made by some leading manufacturers, together with the dimensions of the cylinder and stroke.

Different formulæ for the dimensions of crank-

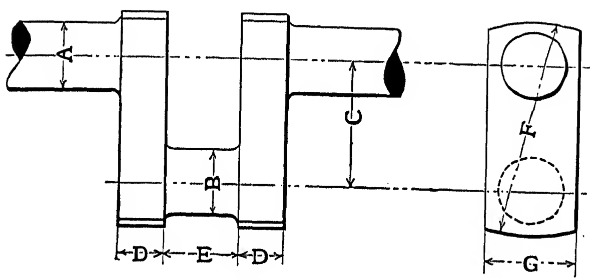


FIG. 10.

shafts are given by various writers on this subject.* The following, for example (which is recommended by the writer), is given by Mr. William Norris.

$$D = \sqrt[3]{\frac{S \times l}{120}}.$$

S = load on piston (area of cylinder in inches × maximum pressure of explosion.

l = length of stroke in feet.

D = diameter of crank-shaft in inches.

*An alternative formula is $D = 0.137\sqrt[3]{S \times l}$.

This formula, however, neglects the bending action due to the distance of the centre of crank-pin from the centre of the bearings. The diameter should be thus slightly increased. Mr. Norris also gives a lengthy description, with example, of ascertaining all the dimensions of the crank-shaft by means of the graphic method.

TABLE I.—SIZES OF CRANK-SHAFTS.

Cylinder.		A.	B.	C.	D.	E.	F.		G.
Diam.	Stroke.	in.	in.	in.	in.	in.	ft	in	in.
5	8	$1\frac{3}{4}$	$1\frac{7}{8}$	4	$1\frac{1}{2}$	2		$6\frac{1}{2}$	$2\frac{1}{2}$
$5\frac{3}{4}$	9	$2\frac{1}{4}$	3	$4\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{5}{8}$		$8\frac{1}{2}$	$3\frac{1}{2}$
$7\frac{1}{2}$	11	$2\frac{3}{4}$	$3\frac{1}{4}$	$5\frac{1}{2}$	$2\frac{3}{8}$	3		$9\frac{1}{2}$	$4\frac{1}{2}$
$8\frac{1}{2}$	15	$3\frac{1}{4}$	4	$7\frac{1}{2}$	$2\frac{7}{8}$	$3\frac{1}{2}$		$12\frac{1}{2}$	5
$8\frac{1}{2}$	18	$3\frac{1}{4}$	4	9	3	$3\frac{1}{4}$	1	2	5
$9\frac{1}{2}$	18	$3\frac{1}{2}$	$4\frac{1}{4}$	9	$3\frac{3}{8}$	$3\frac{1}{4}$	1	3	$5\frac{1}{2}$
12	18	$4\frac{1}{4}$	$4\frac{3}{4}$	9	$3\frac{3}{4}$	$4\frac{1}{2}$	1	$3\frac{1}{4}$	$6\frac{1}{4}$
$11\frac{1}{2}$	21	$4\frac{1}{4}$	$4\frac{3}{4}$	$10\frac{1}{2}$	4	$3\frac{1}{2}$	1	$3\frac{1}{2}$	$6\frac{1}{2}$
14	21	$5\frac{1}{2}$	$5\frac{3}{8}$	$10\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{4}$	1	5	$8\frac{1}{2}$
17	24	7	8	12	$5\frac{3}{4}$	$7\frac{1}{2}$	1	$10\frac{1}{2}$	10
19	30	$7\frac{1}{2}$	8	13	6	9	2	2	11
7	12	$2\frac{7}{16}$	$2\frac{5}{8}$	6	$2\frac{3}{16}$	$2\frac{5}{8}$		$8\frac{3}{4}$	$3\frac{3}{4}$
9	14	$2\frac{15}{16}$	3	7	$2\frac{1}{4}$	$3\frac{3}{8}$			4
11	15	$3\frac{7}{16}$	4	$7\frac{1}{2}$	$2\frac{9}{16}$	$4\frac{1}{8}$		$12\frac{1}{8}$	$4\frac{5}{8}$
$13\frac{1}{2}$	16	$3\frac{15}{16}$	$4\frac{7}{8}$	8	$3\frac{9}{16}$	$4\frac{7}{8}$		$13\frac{3}{8}$	$5\frac{3}{8}$

THE BALANCING of crank-shafts and reciprocating parts is another important feature of an oil engine. With a single-cylinder explosive engine to perfectly accomplish the balancing is impracticable. Most manufacturers, therefore, only balance their engines as far

as the horizontal movement is concerned. The following formulæ is considered correct, and has proved satisfactory for the horizontal type of engines:

$$w = \frac{(C \times R) + G + (S \times r)}{a}.$$

w = weight in lbs. of balance weight.
C = crank-pin and rotating part of connecting-rod in lbs.
R = radius of crank circle in inches.
G = two-thirds weight of all remaining reciprocating parts in lbs.
S = weight of crank-arms in lbs.
r = distance of centre of gravity of crank-arms from centre of rotation.
a = distance of centre of gravity of counterweight from centre of rotation.

Some designers, however, the writer has observed, make the crank balance weights as large as space between bearings and engine bed will allow—that is, when the weights are fastened to the crank-arms, as shown in Fig. 11, thus overbalancing the crank and reciprocating parts. While this would appear bad practice, such engines have been known to run without the slightest vibration. For the vertical type of engines the whole weight of the reciprocating parts, instead of two-thirds weight, has been satisfactorily taken.

Reciprocating parts are sometimes balanced by recess in fly-wheel rim or metal added to the fly-wheel rim or hub. The only correct method of balancing is by counterweights. See Fig. 11.

Various methods of attaching the counterweights to the crank-shaft are shown at Fig. 11, from which it will be noted that the counterweights are attached by studs placed in the cheek of the crank and either pass completely through the counterweight or the counterweight is recessed, the nuts of the studs being tightened in the recess as shown. Again one bolt only is sometimes used, the cheek of the crank-shaft then being recessed, a lip being machined on the counterweight to fit the recess in the cheek of the crank-shaft. The fourth method of attaching the counterweights is shown, in which a bolt is placed at right angles to the center line of the countershaft, this bolt passing through a hole drilled in the counterweights and cheek of the crank-shaft.

The two last named methods are chiefly used in the larger sized engines. The strength of the bolts necessary to hold the counterweights in place can be found by the following formula:

$$d = \frac{n}{13,020}\sqrt[2]{w\,r} + \tfrac{1}{8}''.$$

Where w = weight of one counterweight in lbs.

r = distance from center line of shaft to center of gravity of counterweight in inches.

n = revolutions per minute.

d = diameter of each bolt in inches.

The above is for two bolts for each weight. If one bolt only is used it must equal in tensile strength the two bolts.

CONNECTING-RODS are made of various designs in cross-section, but that chiefly used is made of soft steel and circular, with marine type brasses at crank-pin end and similar bearings at the piston or small end. By some makers the latter bearing is made with adjustable wedge and screw, the end of the connecting-rod then being slotted out, with brass bushes fitted in it.

Each type of connecting-rod is shown at Fig. 12. That illustrated at "A" is a design more suitable for the larger size engines, in which space inside the piston is available for adjustment of the bolts, as shown. The connecting-rod marked "B" is of the rectangular type, and is frequently left rough, the ends only being machined.

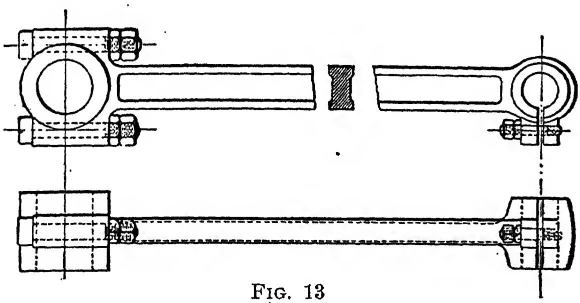


FIG. 13

For small engines a good and cheap form of connecting-rod is made of phosphor-bronze metal, as shown in Fig 13, from which it will be seen that the piston-end bearing is made in one piece with the rod, and being slotted is thus made adjustable. The metal is left rough other than at the bearings.

CONNECTING-ROD BOLTS.—The connecting-rod bolts

should be made of the best wrought iron. The cross-section of connecting-rod bolts at bottom of threads must be such that on the beginning of the suction stroke the stress does not exceed 4,000 to 6,000 lbs. per square inch. The total force is made up of the inertia force and the suction force and is arrived at as follows:

Let $F=$ total inertia force.
$d=$ diameter of cylinder in inches.
$W=$ total weight of piston, piston pin, one-half the weight of connecting-rod and the weight of any cooling water in the piston.
$r=$ radius of crank in feet.
$l=$ length of connecting-rod in feet.

$$\text{Then } F = .00034\,W(\text{R. P. M.})^2 r\left(1+\frac{r}{l}\right),$$

and the suction force = about 2 lbs. per square inch. Therefore the total suction force

$$A = 2 \times .785\,d^2.$$

The area of all the connecting-rod bolts at the root of the threads should not be less than $\frac{F+A}{6,000}$.

The connecting-rod of a single-acting engine has, chiefly, compression stresses to withstand; both the outer end bearings have little or no strain on them, except that due to momentum of the reciprocating parts. The connecting-rod should be from two to three strokes in length. In computing its strength, the connecting-rod can be taken as a strut loaded at either end. The mean diameter when made of mild

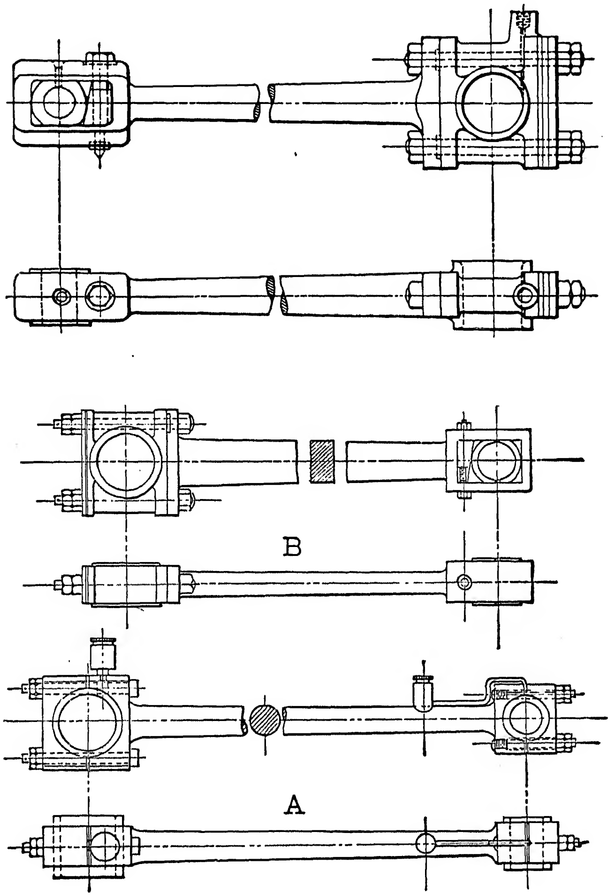


FIG. 12.

should be made of the best wrought iron. The cross-section of connecting-rod bolts at bottom of threads must be such that on the beginning of the suction stroke the stress does not exceed 4,000 to 6,000 lbs. per square inch. The total force is made up of the inertia force and the suction force and is arrived at as follows:

Let F = total inertia force.
d = diameter of cylinder in inches.
W = total weight of piston, piston pin, one-half the weight of connecting-rod and the weight of any cooling water in the piston.
r = radius of crank in feet.
l = length of connecting-rod in feet.

$$\text{Then } F = .00034\, W\,(\text{R. P. M.})^2 r\left(1 + \frac{r}{l}\right),$$

and the suction force = about 2 lbs. per square inch. Therefore the total suction force

$$A = 2 \times .785\, d^2.$$

The area of all the connecting-rod bolts at the root of the threads should not be less than $\dfrac{F+A}{6,000}$.

The connecting-rod of a single-acting engine has, chiefly, compression stresses to withstand; both the outer end bearings have little or no strain on them, except that due to momentum of the reciprocating parts. The connecting-rod should be from two to three strokes in length. In computing its strength, the connecting-rod can be taken as a strut loaded at either end. The mean diameter when made of mild

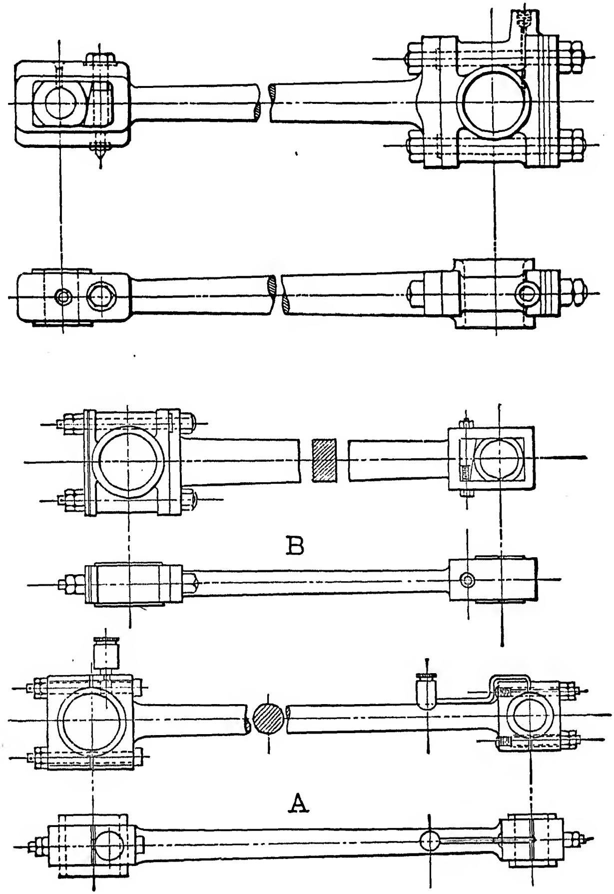


FIG. 12.

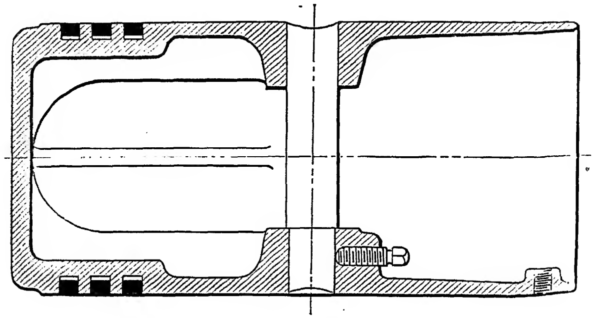


FIG. 14.

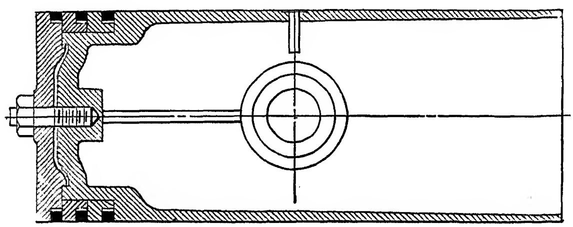


FIG. 14a.

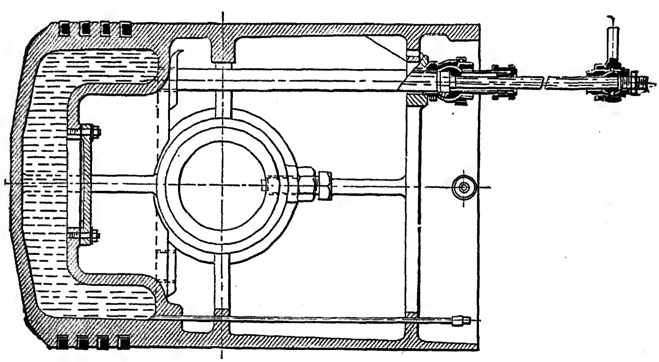


FIG. 15.

steel is arrived at by the following formulæ, as given by authorities on steam-engine design:

$$x = 0.035 \sqrt{D\, l \sqrt{m}}.$$

x = mean diameter of connecting-rod (half sum of diameter of both ends).
D = diameter of cylinder in inches.
l = distance in inches between centre of connecting-rod.
m = maximum explosive pressure in lbs. per square inch.

This formula, however, is excessive for medium and slow speed engines, and in such instances the writer has used the following formulæ with satisfactory results—namely:

$$0.028 \sqrt{D\, l \sqrt{m}}.$$

THE PISTON in single-acting engines is generally of the trunk pattern, as shown in Fig. 14, with internal gudgeon-pin placed in the centre of the piston, secured at either end to the piston by set-screws. The steam-engine cross-head and slide-bars are dispensed with, the power being transmitted directly from the gudgeon-pin of the piston to the crank.

The piston is made of hard close-grained iron, and should not be less than 5-16″ in thickness for small engines and slightly heavier for the larger sizes. In

each case the metal is thicker at the back than at the front end. The piston is usually 1.6 diameters in length. Three cast-iron piston-rings, as shown in Fig. 15, are fitted to the smaller engines, four and five rings being required to keep the piston tight in the larger sizes. A single ring is sometimes added, placed in front of the gudgeon-pin, but its use is not recommended. The pressure on the piston, caused by the explosive pressure and due to the angularity of the connecting-rod, should not be greater than 25 lbs. per square inch of rubbing surface.

The piston in which separate distance-pieces between each ring and having separate plate bolted to the back of the piston is shown at Fig. 14*a*.

In the larger engines (those having a cylinder diameter of more than 24 inches), a water-jacketed chamber is made at the back end of the piston which is supplied with a continuous flow of cooling water. This piston is shown in section at Fig. 15 and Fig. 95. The cooling water is conducted to and fro by separate pipes attached to the piston, as shown in the illustration Fig. 95, and communicate either through stuffing boxes or other suitable means to allow proper supply of water to the piston. Water-jacketing of the piston is necessary in the larger sizes because of the increased volume of burning gases which would become unduly heated, allowing increased expansion of the piston and rendering it difficult of lubrication.

Piston Speed.—The revolutions per minute at which the engine is designed to run is governed almost entirely by the piston speed. High speed engines are designed with a comparatively short stroke—slow speed

engines having a stroke much longer in comparison with the diameter of the cylinder. The maximum allowable speed of the piston is considered as 900 feet per minute. As in four-cycle engines the operation of the valves takes place only every other revolution, this type of engine is made with a speed frequently as high as 350 to 400 R. P. M.

Inertia force per square inch of piston at end of compression stroke must not exceed compression pressure, or the explosion will reverse the direction of pressures and cause a "knock."

The inertia force per square inch of piston $\frac{F}{a}$ will be as follows:

$$\frac{F}{a} = \frac{.00034\, W\,(\text{R. P. M.})^2}{a} r \left(1 + \frac{r}{l}\right).$$

a = area of piston in sq. in.

The value of $\frac{F}{a}$ must be such as to be less than the compression pressure.

FLY-WHEELS.—The oil engine is equipped with heavier fly-wheels than is necessary with a steam engine. The weight of the oil engine fly-wheel varies inversely both with the number of impulses given per revolution at the crank-pin and the degree of unsteadiness from the uniform speed of rotation allowed. The total revolutions per minute are controlled by the governor, but the cyclic variation and the degree of unsteadiness from uniform speed of rotation during one cycle depend on the fly-wheel. For a given degree of unsteadiness of a single cylinder, single acting four-cycle engine, the heaviest fly-wheel will be required.

Where the number of cylinders is increased, or where the number of impulses per minute are increased, the weight of the fly-wheel to give the same degree of unsteadiness will, of course, be less than with a single cylinder engine previously referred to.

By the degree of unsteadiness is meant the change in speed from the uniform speed of rotation throughout the cycle.

Let T = Degree of unsteadiness.

Then $$T = \frac{V\,\text{max} - V\,\text{min}}{V\,\text{ave}}.$$

V max = maximum velocity of shaft during cycle.
V min = minimum velocity of shaft during cycle.
V ave = average velocity of shaft during cycle.

The value of T recommended by Güldner* is:

.05 to .0334	$\frac{1}{20}$ to $\frac{1}{30}$	for pumps and wood factories.
.0285 to .025	$\frac{1}{35}$ to $\frac{1}{40}$	for factories.
.025	$\frac{1}{40}$	for looms and paper mills.
.020	$\frac{1}{50}$	for grinding mills.
.0166 to .001	$\frac{1}{60}$ to $\frac{1}{100}$	for spinning factories.
.00067	$\frac{1}{150}$	for direct-current generator.
.00033	$\frac{1}{300}$	for alternating-current generators.

By cyclic variation is meant the greatest angle that the rotating crank-pin varies from the position it would occupy were its motion perfectly uniform. Generally these two conditions are not related. Consideration of

*Verbrennungs motoren H. Güldner. Page 345.

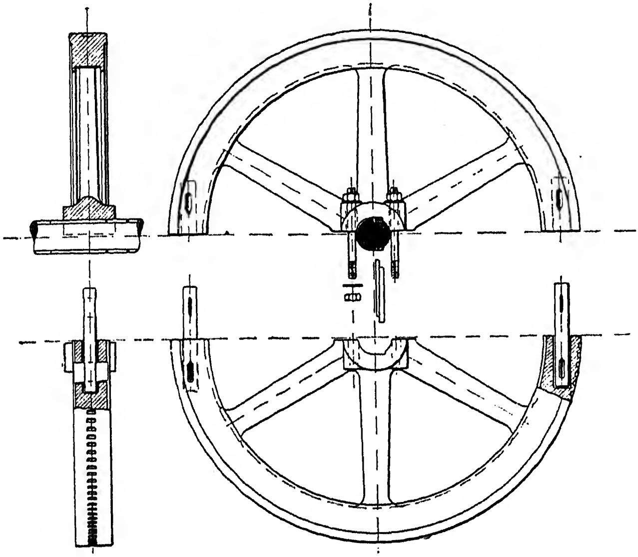


Fig. 16a

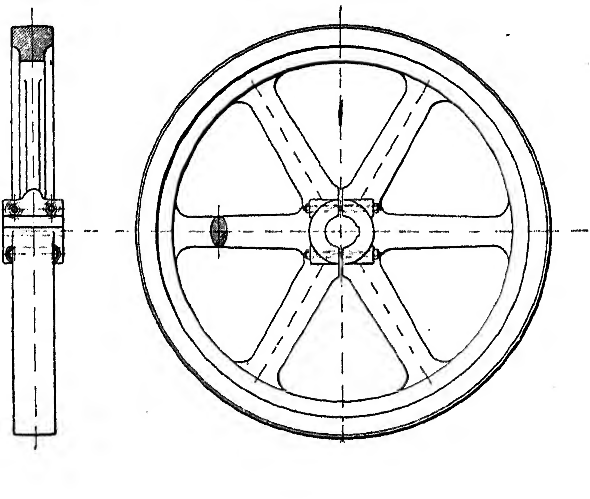


Fig. 16

Where the number of cylinders is increased, or where the number of impulses per minute are increased, the weight of the fly-wheel to give the same degree of unsteadiness will, of course, be less than with a single cylinder engine previously referred to.

By the degree of unsteadiness is meant the change in speed from the uniform speed of rotation throughout the cycle.

Let T = Degree of unsteadiness.

Then $$T = \frac{V \max - V \min}{V \text{ ave}}.$$

V max = maximum velocity of shaft during cycle.
V min = minimum velocity of shaft during cycle.
V ave = average velocity of shaft during cycle.

The value of T recommended by Güldner* is:

.05 to	.0334	$\frac{1}{20}$ to $\frac{1}{30}$	for pumps and wood factories.
.0285 to	.025	$\frac{1}{35}$ to $\frac{1}{40}$	for factories.
	.025	$\frac{1}{40}$	for looms and paper mills.
	.020	$\frac{1}{50}$	for grinding mills.
.0166 to	.001	$\frac{1}{60}$ to $\frac{1}{100}$	for spinning factories.
	.00067	$\frac{1}{150}$	for direct-current generator.
	.00033	$\frac{1}{300}$	for alternating-current generators.

By cyclic variation is meant the greatest angle that the rotating crank-pin varies from the position it would occupy were its motion perfectly uniform. Generally these two conditions are not related. Consideration of

*Verbrennungs motoren H. Güldner. Page 345.

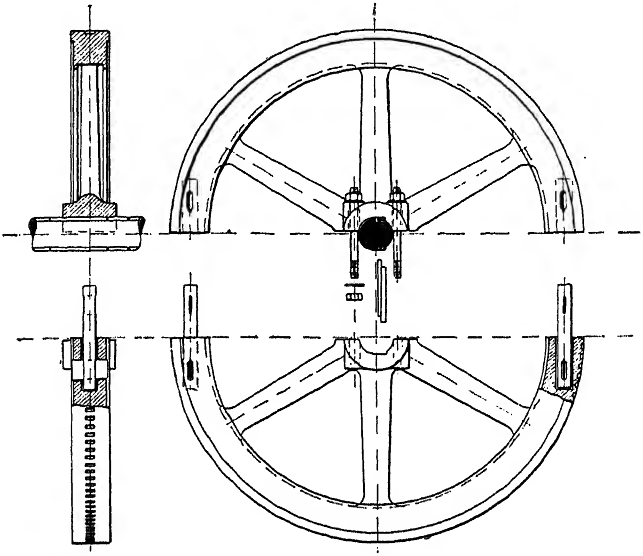


Fig. 16*a*

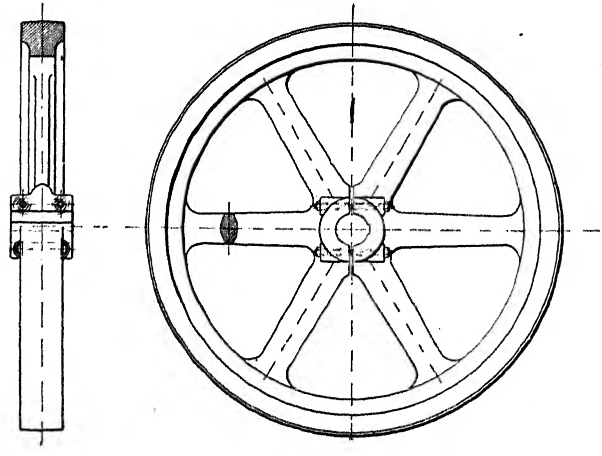


Fig. 16

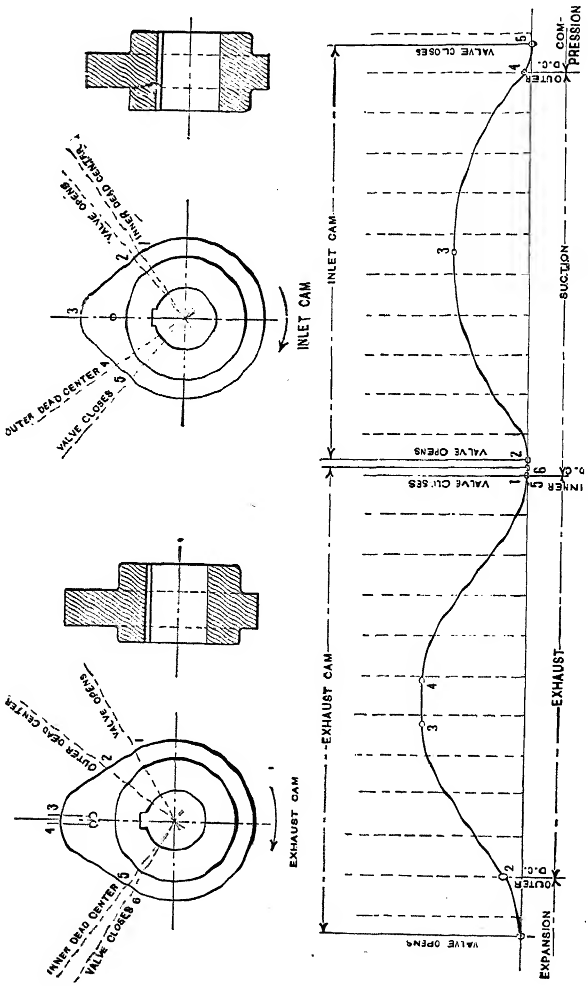

INNER DEAD CENTER
VALVE OPENS
OUTER DEAD CENTER 4
VALVE CLOSES 5
INLET CAM
OUTER DEAD CENTER
VALVE OPENS
INNER DEAD CENTER
VALVE CLOSES 6
EXHAUST CAM
COM-
PRESSION
OUTER D.C.
SUCTION
INNER D.C.
EXHAUST
EXPANSION
VALVE CLOSES
VALVE OPENS
VALVE CLOSES
VALVE OPENS

FIG. 17.

cyclic variation is usually only necessitated when the engine is required to operate alternators in parallel or where a similar uniform motion is necessary.

The diameter of the fly-wheel is governed by the peripheral speed which should not exceed 6,000 ft. per min. for cast iron. In computing the weight of the fly-wheel, it is customary to neglect the weight of the hub and arms, and to calculate only on the weight of the rim as follows:

$W =$ weight of rim only in tons (2,000 lbs.).
$D =$ dia. of the center of gravity of rim in feet.
$N =$ revolutions per minute.
$P =$ actual or brake H. P.
$C =$ constant.

Then $$W = C \frac{P}{D^2 T N^3}.$$

$C =$ for single-acting 4-cycle engine with impulse each 720°, 520.000.
$=$ for engines with impulse each 360°, 250.000.
$=$ for engines with impulse each 240°, 166.000.
$=$ for engines with impulse each 180°, 83.000.

Different types of fly-wheels are shown at Fig. 16. The smaller engines for industrial purposes are equipped with one and sometimes two fly-wheels made in one piece. Larger engines of say 50 H. P. and upwards are usually equipped with one large fly-wheel made in two parts as shown at Fig. 16*a*. The hub split with medium sized wheels is considered advantageous, as it allows more accurate fitting to the shaft and it becomes easier to keep the wheel running in truth.

The cams are made of cast iron or steel and are usually designed as shown in Fig. 17. Cast iron is ad-

vantageously chilled to withstand the wear of the rollers.

The function of a cam is to transfer rotary motion of the crank-shaft and cam-shaft to the reciprocating action required for lifting the poppet valves. The rapid opening and closing of the valves necessary in a four-cycle engine is more easily arrived at with a cam motion than otherwise. The valve is closed by a spring, the function of opening the valve being performed by the cam only. Generally valve mechanisms in which cams and poppet valves are used are noisy in operation, especially in higher speed engines.

The rate of opening and closing of the valve can be ascertained by plotting a curve corresponding to ordinates equivalent to the various distances from the face of the cam to its centre taken at specified intervals. The required width of the face of the cam in contact with the rollers is ascertained by computing the work to be done due to the pressure in the cylinder at time of valve opening, together with the area of the valve. Accordingly, where the air valve is operated the cam controlling its movement is of less width, seeing that only atmospheric pressure obtains when it is operated as compared with the exhaust valve cam, which has to open that valve against a pressure in some cases as high as 40 lbs., necessarily involving considerable work.

VALVES AND VALVE-BOXES.—The dimensions of the air-inlet and exhaust valves are governed by the diameter of the cylinder and the piston speed. The style of the valve-box recommended is that made separate and bolted to the cylinder. The valve-box can then

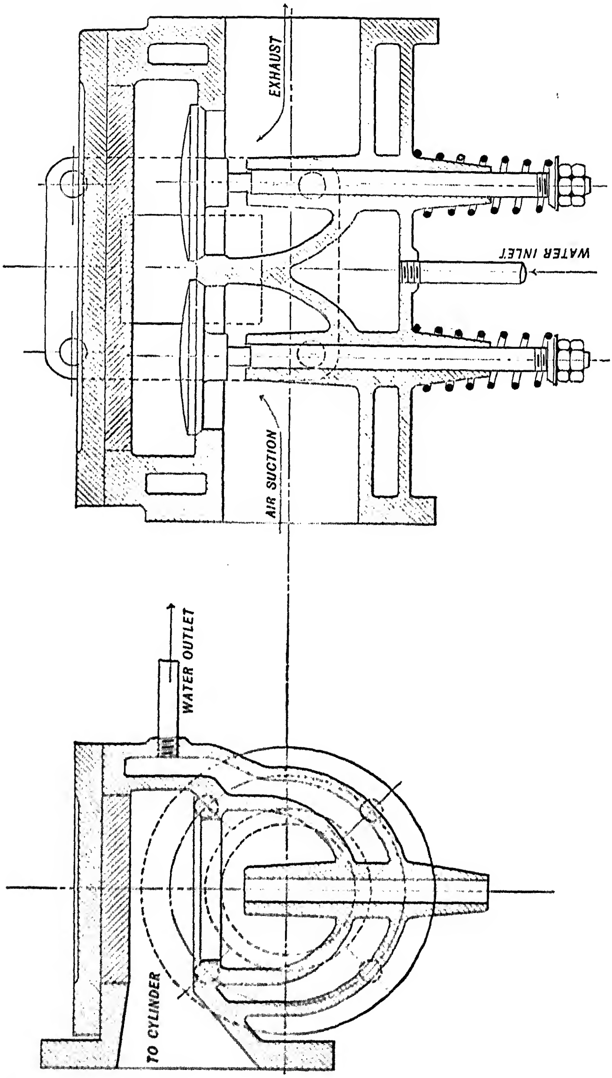


Fig. 18.

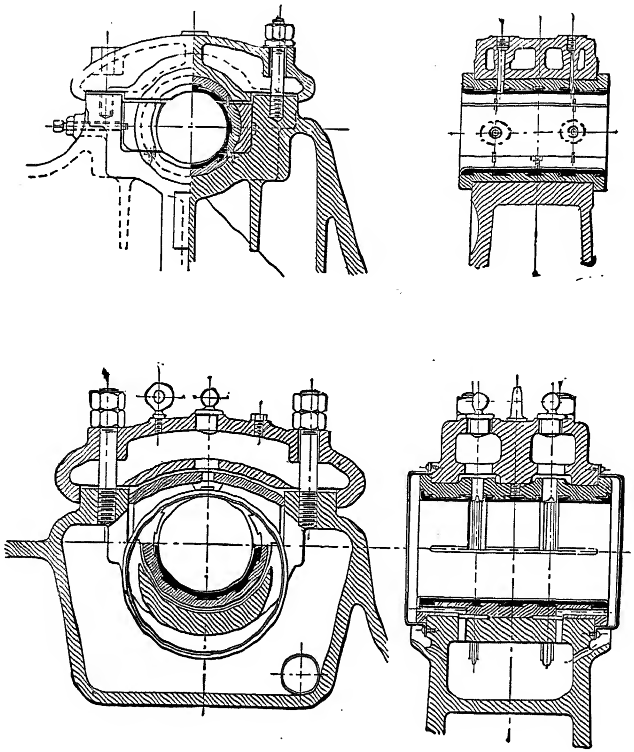


FIG. 18*a*.

be entirely renewed if necessary and at small expense. This type of valve-box is shown at Fig. 18, both valves being operated from the cam-shaft. The springs required to close the valves are shown at Figs. 18 and 19. The latter arrangement has the advantage of having the springs placed away from the heated valve chambers. Other designs of valve chambers have the valves placed horizontally in the cylinder back-head. A compact design of valves is shown at Fig. 20, in which the exhaust valve is operated only, the air valve being automatic. In each case the valves should be brought as close as possible to the cylinder walls, the clearance space in the ports, etc., being reduced to a minimum.

With engines of larger size the air and exhaust valve box is surrounded by a water jacket, which maintains its proper temperature and prevents the seats of the valves being distorted by undue expansion, which might otherwise occur. It will be noted in the illustration that the inlet and outlet water connections to the valve-box are made by separate pipes.

Where the air-inlet valve is made automatic, it is opened by the partial vacuum in the cylinder during the suction period, and closed by a delicate spring, as shown in Fig. 20. The air and exhaust valves and port openings are usually made of such an area that the velocity of the air inlet as it enters the cylinder is 100 feet per second—the velocity of the exhaust gases through the exhaust or outlet being about 80 feet per second, presuming the exhaust products to be expelled at atmospheric pressure. The air-inlet valve, if automatic, should be so arranged as to allow ingress of air

without choking. In calculating the area of valve ports or passages, allowance must be made for valve guide or other obstruction in the passages. The velocity of the air is found in the following formulæ:

$$V = \frac{a \times P}{a_1}$$

$V =$ velocity of air in ft. per second.
$P =$ piston speed in ft. per second.
$a =$ area of piston in inches.
$a_1 =$ area of valve opening in inches.

Main Bearing.—Various designs of bearings are shown at Fig. 18*a*. The ring oiling type of bearing, while somewhat more expensive to manufacture than the other types shown, is recommended. The maximum pressure on the bearing should not exceed 400 lbs. per sq. in. of projected area.

The Crank-pin.—To determine the dimension of the crank-pin would properly lead to a lengthy discussion as to all the strains involved, and the reader for a complete discussion on this subject is referred to works where space is allowed for such.*

In different types of engines the dimension of the pin varies. A crank-pin short in length and comparatively large in diameter is recommended. The diameter of the pin being not less than 1.2 times the diameter of the shaft. (See table I.)

The average pressure on the crank-pin allowable should not exceed 500 lbs. per sq. in. of projected area.

*Unwin Machine Design.

THE EXHAUST BENDS close to valve-box should when possible be of not less than 5″ radius for the smaller engines, which dimension should be increased for larger-sized engines.

THE VALVES are made of forged steel, either in one piece or with cast-iron valve and wrought-iron or steel stem fitted into it, and are shown in Fig. 21. Some manufacturers prefer the latter on account of cheapness, and also because it is claimed the cast-iron valves will withstand heat better than the forged valve.

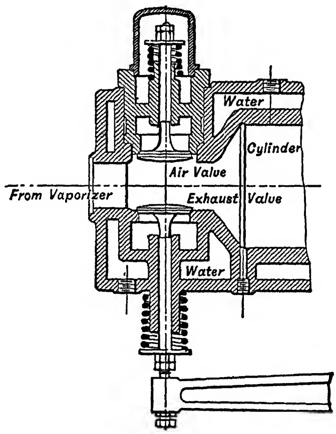


FIG. 20.

PISTON-PIN.—For small engines, the length of the piston-pin is almost invariably one-half the diameter of the cylinder and the diameter of the pin 0.15 to 0.25 the diameter of the cylinder. This leads to pressures of 1,800 to 2,200 lbs. per sq. in. of projected area.

Medium power and large engines have piston-pins of diameter

minimum $dp = 0.22d$ where $d =$ diameter of cylinder.
maximum $dp = 0.45d$.
mean $dp = 0.31d$.

Lucke recommends a pin of diameter*

$$dp = 0.35d$$

and of length

$$lp = 0.6dp.$$

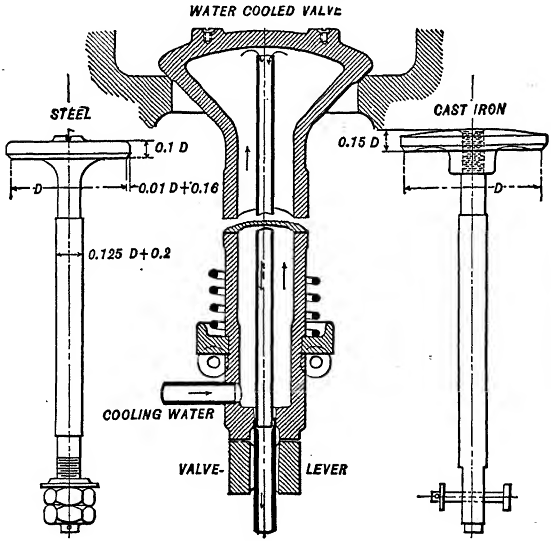


FIG. 21.

THE ENGINE FRAME.—Different designs of engine frames are shown in the illustrations of sectional views

*Gas Engine Design by C. E. Lucke, Ph.D.

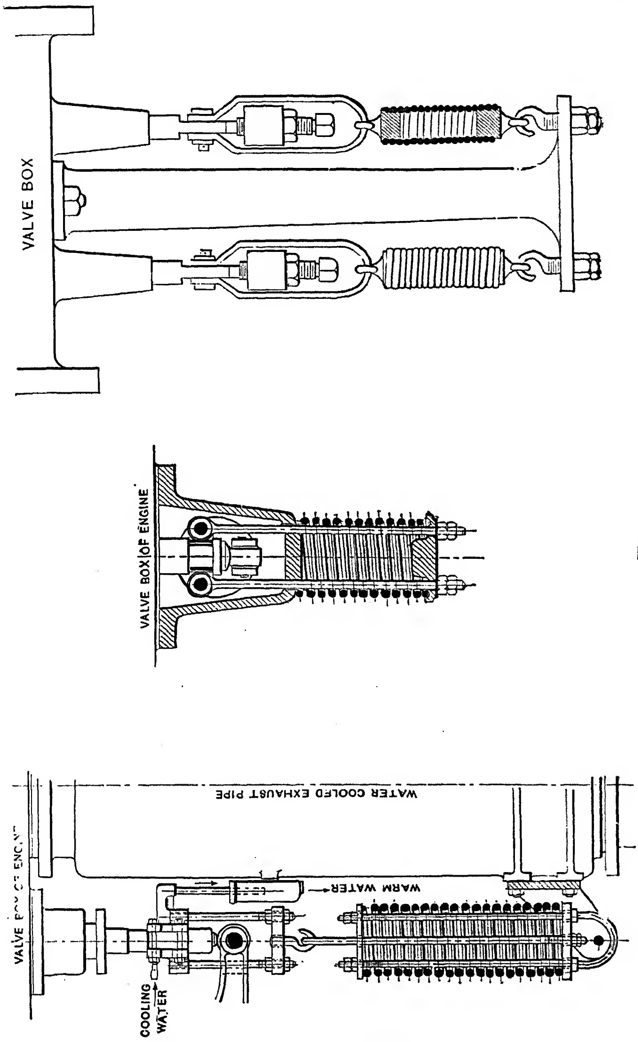

VALVE BOX
VALVE BOX OF ENGINE
WATER COOLED EXHAUST PIPE
WARM WATER
COOLING WATER

FIG. 19.

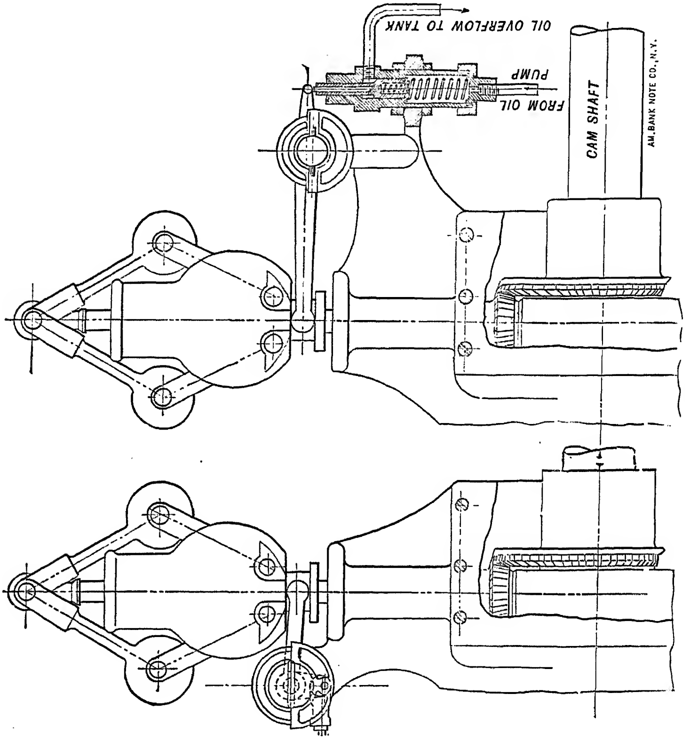


FIG. 24.

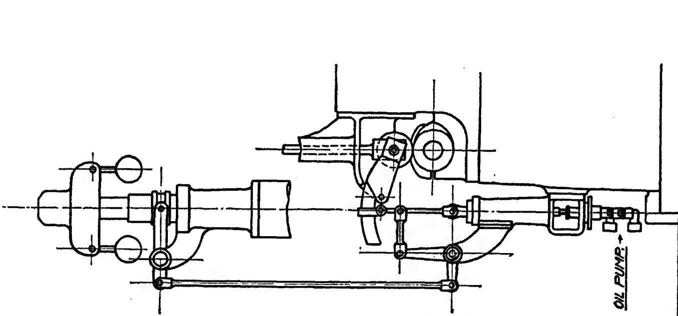


FIG. 24a.

of various engines (see Figs. 76, 98, 110). The frame should be proportioned not only to prevent vibration and to withstand the strains consequent on the impulse in the cylinder, but should also be ribbed and of ample sectional strength to overcome the vibration known as "panting."

VALVE MECHANISMS.—With the Beau de Rochas or four-cycle engine the valves are only operated during alternate revolutions of the crank-shaft. This necessitates an arrangement of some kind of two-to-one gear. Worm-gear, as shown in Fig. 22, is considered

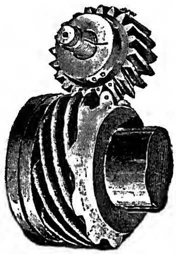


FIG. 22.

to be well adapted for this work. The power necessary to operate the valves is, in this case, transmitted from the crank-shaft by the worm or skew gearing through the cam-shaft, with separate cams opening the air and exhaust valves by the operating levers, as shown in Fig. 23. Where spur-gearing (Fig. 23*a*) is used the cam-shaft is mounted in bearings parallel to the crank-shaft, the cams then acting on the horizontal rod working in compression, which opens the valves.

Various other arrangements for reducing the motion are also used, the work accomplished being in each

case the same as with the worm or spur gear, shaft and levers—namely, the opening of the valves during alternate revolutions of the crank-shaft.

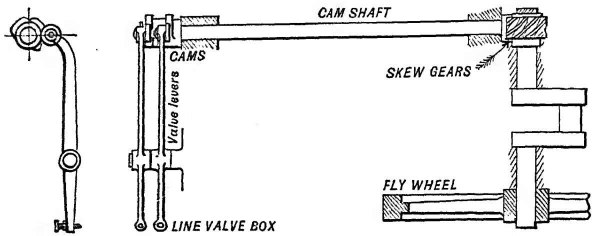


FIG. 23.

In the two-cycle engine this valve or valves are operated each revolution of the crank-shaft by eccentric or cams actuated directly from the crank-shaft.

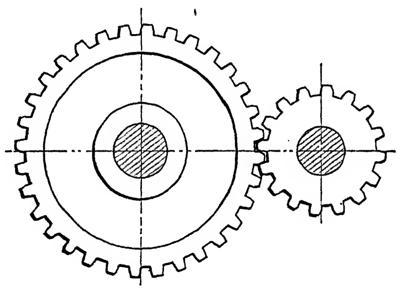


FIG. 23*a*.

GOVERNING DEVICES.—The governing devices for controlling the speed of oil engines are of two kinds: first, that designed to develop centrifugal force, which

is balanced either by suitable controlling spring or dead weight, as shown in Fig. 24, and, secondly, the inertia or pendulum type of governor.

The accompanying illustrations also show the method of by-passing the oil where the air supply is constant at all loads. The Rites governor, a very simple and efficient device of the fly-wheel type of governor, is illustrated and described in Chapter X., the method of governing, in which the air supply and oil supply is controlled, is shown at Fig. 7, illustrating the Priestman governor. In those engines where the regulation is controlled by preventing the suction into the cylinder, caused by holding the exhaust valve open, the inertia type of governor is sometimes used, where the inertia of a weight attached to a reciprocating part of the valve motion is arranged, having its movement controlled by an adjustable spring. When the normal speed is exceeded the inertia of the weight overcomes the pressure of the spring and thus holds open the exhaust valve till the normal speed is regained.

The governors regulate the speed of the engine by the following different methods:

(*a*) By acting through suitable levers or other mechanism on the valves controlling the fuel supply to the cylinder, either by means of a by-pass valve placed in the oil-supply pipe to vaporizer, thus allowing part of the charge of oil to return to the tank instead of entering the vaporizing chamber or by regulating the amount of oil as well as the air supply.

(*b*) Acting directly on the oil-supply pump, length-

ening or shortening the stroke of the pump, as required.

(*c*) Where the oil vapor is arranged to be drawn into the cylinder with the incoming air the governor

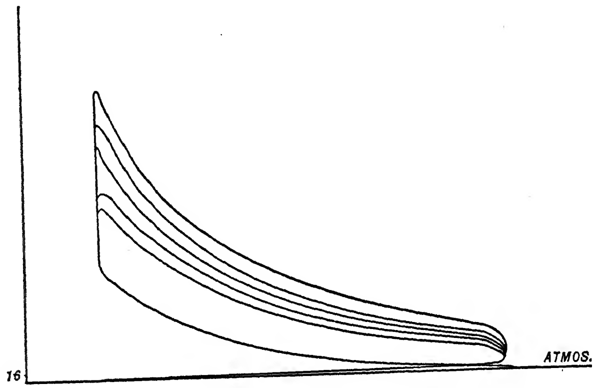


FIG. 25.

acts on the exhaust-valve, holding it open during the suction stroke, thus preventing the inlet of vapor to the cylinder.

(*d*) By acting on the vapor inlet-valve, allowing this valve to open only when an impulse to the piston is required.

Engines driving dynamos for electric lighting and requiring very close regulation are preferably governed by the system of throttling or reducing the explosive pressures in the cylinder. Thus, when the engine exceeds the standard speed for which the governor is set, only part of the vapor or oil is allowed to enter the

vaporizing chamber or cylinder. The mixture of oil,

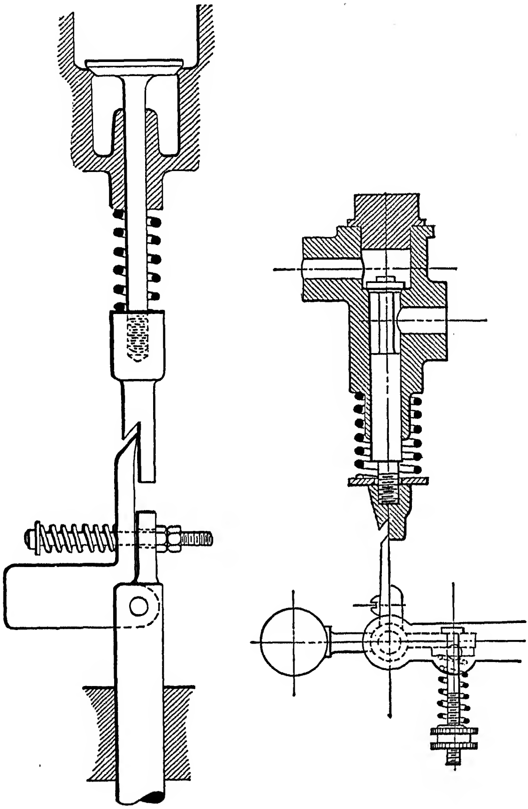


FIG. 26.

vapor and air is accordingly regulated, and the mean effective pressure as required is suitably reduced.

The indicator diagram illustrates the variation of the M. E. P. in the cylinder, as shown in Fig. 25, each expansion line registering a different pressure. No explosion is in this case omitted entirely, and consequently the running of the engine is even and regular. A governor acting directly on the oil supply pump is shown at Fig. 24*a*. Another type of governor operating on the fuel oil pump directly is shown at Fig. 24*b*. In this instance the governor is placed within the fly-wheel and is also arranged to operate directly on the oil pump. It consists of frame *F* fastened concentrically to inside of flywheel cam ring *R*, which has projection *B* and cam *C* projecting and operating each revolution (with 2-cycle type) on roller *A*, causing movement of plunger *P*. *W* is a wedge on lever *L* which separates *R* from *F*. If the speed is increased above normal the counterweight *H* overcomes the tension of spring *S*, moving the wedge outwards, allowing the buffer *G* to move from plunger *P*; thus the lift of *C* is reduced and the length of pump stroke reduced.

The hit-and-miss type of governor is shown in Fig. 26. This device is made in many different forms, the mode of working being similar in them all—namely, the inertia of a weight controlled by the spring. When the speed of the crank-shaft is increased the weight is moved correspondingly quicker; its inertia is then increased, and the strength of the spring is overcome sufficiently to allow the engaging parts of the valve

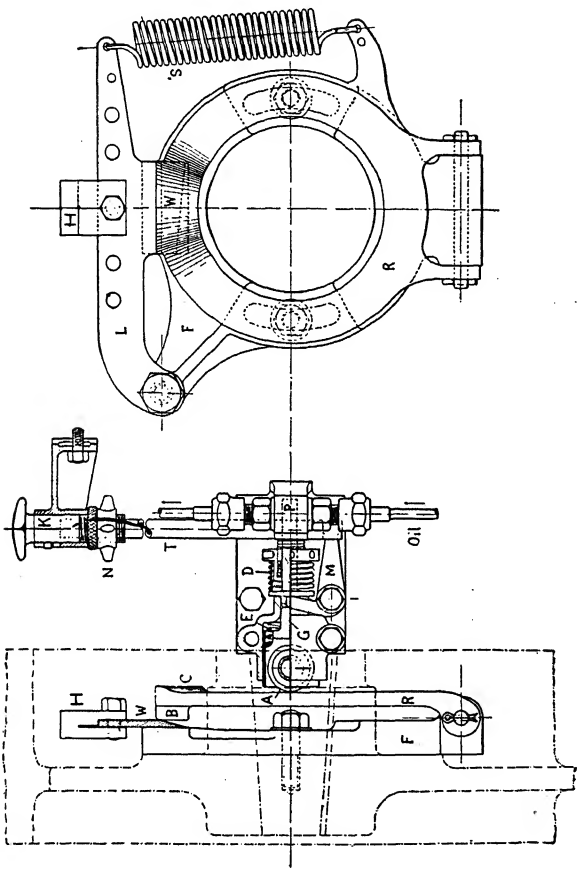


FIG. 24*b*.

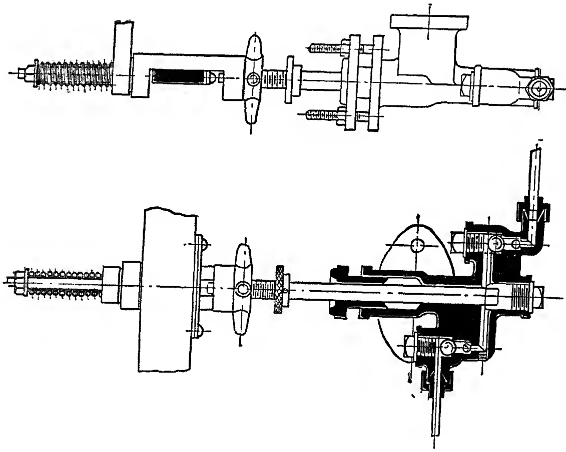


FIG. 28.

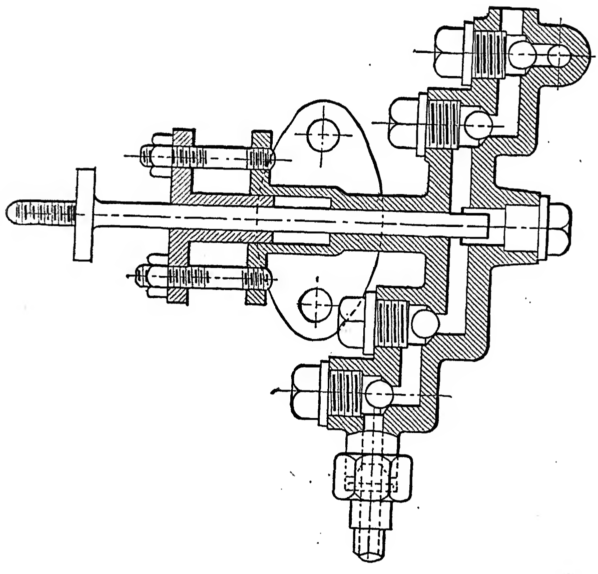


FIG. 27.

motion to be disengaged during one or more revolutions, and consequently where this device acts on the oil-pump the charge of oil is missed, and no explosion takes place during the following cycle of operations.

THE OIL-SUPPLY PUMP is placed against the oil-tank and base of engine or on bracket bolted to cylinder. It is usually made of bronze, with steel ball valves. Duplicate suction and discharge valves are advantageous in case one valve on either side should leak. Figs. 27 and 28 represent oil-pumps as used on the Hornsby-Akroyd oil engine.

THE FUEL OIL-TANK is placed in or bolted against

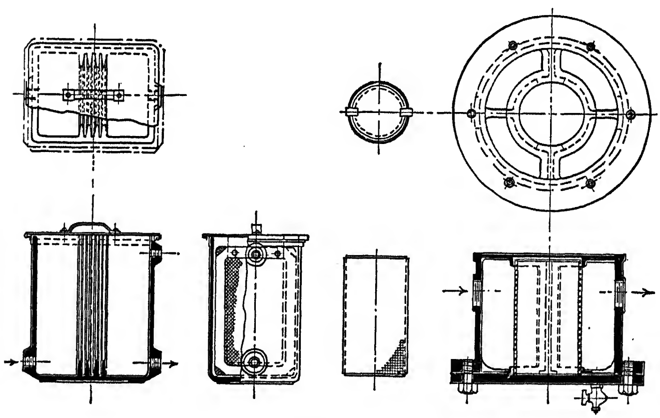


FIG. 29.

the base of the engine. It is then made of cast iron as part of the base of the engine; otherwise the tank is made of galvanized iron and separate from the engine

base, so that it can be taken out when required for cleaning.

A filter or strainer for cleaning the oil as it passes to the oil-pump which can be placed where convenient and is separate from the oil tank is shown at Fig. 29.

Horizontal as Compared with the Vertical Type of Oil Engines.

The accessibility of the piston with the horizontal engine is considered a great advantage. The piston can always be seen and can be drawn out of the cylinder and cleaned and replaced with ease in this style of engine, whereas in a vertical engine it is necessary to remove the cylinder cover, and perhaps other parts, to gain access to the piston, and also it is necessary to have sufficient head room above the top of the cylinder for chain-block to lift the piston and connecting-rod. The lubrication of the piston is also considered more effective in the horizontal than in the vertical type of engine. Furthermore, the connecting-rod is more accessible for adjustment both at the crank-pin end and at the piston end in the horizontal type. This difficulty, however, has been overcome by arranging a removable plug in the cylinder casing, which when taken out allows access for adjustment to the piston end of the connecting-rod. European designers seem much in favor of the horizontal type of engines, and although some leading makers build the vertical type of engines, yet the greater number would appear to be made of the horizontal type.

Vertical engines for situations in buildings where space is restricted and where sufficient head room is available have the great advantage of occupying less floor space than the horizontal type. The mechanical efficiency of a vertical engine is somewhat greater, the friction of the piston being less than in the horizontal type of engine.

The vertical type for some special purposes can, of course, only be used, but for ordinary uses the horizontal type of engine at present seems to be most in favor, one consideration being the difficulty of suitably arranging the vaporizing and spraying details in the vertical type of engine, which are usually placed close to the cylinder, and are, therefore, not so fully under the control of the attendant as in the horizontal type.

Multi-cylinder Engines.—For industrial purposes and situations where simplicity of construction is of prime importance and where the engine will have little or no skilled attention, the single cylinder horizontal engine is preferred on account of fewer moving parts. Objection is frequently made to a multi-cylinder or twin-cylinder engine on this account. The multi-cylinder engine, however, has the advantage that an impulse is received at the crank-pin with greater frequency than is the case with the single cylinder engine. For example, in the single four-cycle engine one impulse is received during two revolutions, while in the two-cycle single cylinder engine one impulse per revolution takes place. With the multi-cylinder engine, for instance, three-cylinder type, four-cycle

single acting, three impulses are received by the crank-pin each two revolutions and with the three-cylinder two-cycle type six impulses in two revolutions. The multi-cylinder engine, therefore, has an important advantage over the single cylinder type for such purposes as electric lighting and especially for operating alternating generators running in parallel where least possible cyclic variation is required.

Again, the multi-cylinder engine has the adavantage, considering that each impulse is more frequent, of not requiring the heavy fly-wheel necessary with the single cylinder type as explained on page 36. Undoubtedly the multi-cylinder type engine requires much more adjustment of bearings than those of the single cylinder type. The multi-cylinder type being lighter in weight per actual horse-power can be manufactured cheaper per horse-power than can the single cylinder type.

WATER INJECTION.—The injection of a small amount of water, water vapor, or steam into the vaporizer or cylinder of the oil engine is now the practice of several makers. In the sectional view of the latest type of Crossley vaporizer, Fig. 3, is shown a water inlet valve to the vaporizer whereby a very small amount of water is injected into the vaporizer as well as air and fuel. The Priestman engine has an arrangement also allowing a small amount of water to be drawn into the combustion chamber when the engine is operating at full load.

The Mietz & Weiss engine is arranged to allow steam formed in the water jacket surrounding the cylinder to enter the combustion chamber with the fuel. The

advantages claimed for the injection of water, etc., are first, that the engine works more quietly with it than without. The heavy blow of the explosion and the metallic knock heard at full load is reduced; and secondly, with the water injection a somewhat higher compression can be used without fear of pre-ignition; and thirdly, the lubrication of the cylinder is assisted and the piston is maintained in a cleaner condition. The chief disadvantage is found when the supply of water is not very carefully regulated. The timing of ignition may be retarded or become irregular if too much water is admitted.

Time of Injection of Fuel.—In the descriptive matter relative to the Diesel engine, page 216, it is pointed out that the injection of the fuel takes place after compression of the air in the cylinder is completed. This was a feature peculiar to this engine. Several other makers are now adopting this feature; that is, increasing the compression and injecting the fuel as (or a few degrees before) the piston reaches the inner dead centre. The increased compression results in increased economy and more complete combustion of the fuel. In the latest type Hornsby oil engines, in the De la Vergne F. H. type, and in the smaller 2-cycle type described in Chapters X. and XII. this feature is referred to.

Erecting and Assembling of Oil Engines.—The following remarks relating to the erection of oil engines contain a few hints on important points of this work, the information being intended for those

readers not sufficiently familiar with the assembling of explosive engines to be cognizant of the parts requiring careful handling and accurate workmanship.

BEARINGS.—In scraping in the crank-shaft bearings of horizontal engines the shaft must bear perfectly on that part of the bearings as shown in Fig. 30, marked

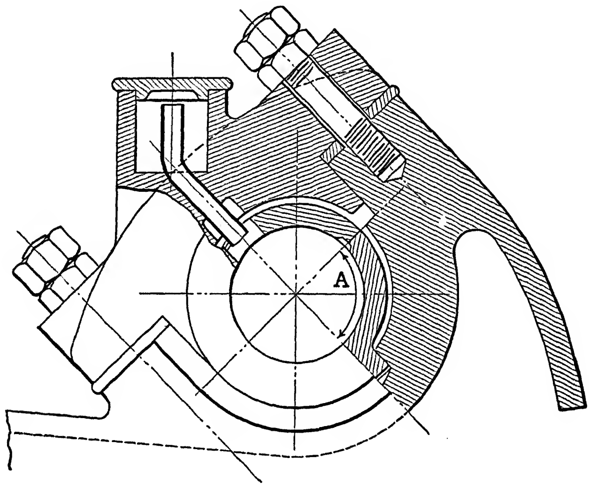


FIG. 30.

A, the greater pressure being on the part of the bearing which is between the centre line of engine drawn through the cylinder and the part through which the vertical centre line of fly-wheel is drawn. A slight play of about 1-64″ can be given to the crank-shaft sideways in the bearings in smaller-sized engines, and 1-32 of an inch in the larger sizes is recommended.

In vertical engines the bearings receive both the pressure of explosion and the pressure due to the weight of the fly-wheels in the same part, and these bearings require the same care at those points in the lower half of the bearing—namely, about 45° each side of the centre line drawn vertically through the cylinder and crank-shaft. The bearing surfaces of the caps and of that part where the pressure is not so great do not require such careful scraping as those parts where the pressure is greater.

Piston and Piston-Rings.—The fitting of piston and piston-rings is very important and requires accurate workmanship. The cylinder and piston are machined to standard ring and gauge, one-thousandth per inch diameter of cylinder play being allowed. The metal of the piston not being of uniform thickness after machining may slightly lose its shape, and sometimes requires slight hand-filing when being fitted to the cylinder. The piston without rings can be moved easily up and down inside the cylinder. If necessary the piston should be eased slightly by hand on the sides, being left a good and close fit at the top and bottom bearing in horizontal engines. The sides should not rub hard in any part. The piston, if the rings are in place, can be fitted to the cylinder from the back end of the cylinder, and can be moved around the front end, being inserted into cylinder as far as the rings.

The distance-pieces or junk-rings should not touch the sides of the cylinder, the bearing of the piston being only on the trunk of the piston itself. The front

part of the piston can also be bevelled for $\frac{3}{4}''$ in length, 1-32″ in diameter, as shown in Fig. 14.

THE PISTON-RINGS, if made as in Fig. 15, should have in the smaller sizes 1-32″ play, in the larger sizes 1-16″, as shown at A in Fig. 31. This space allows for expansion when the ring becomes heated in working. It is advantageous to insert dowel-pins in the piston grooves to maintain the rings in the same position, so that the space in each ring is out of line with that in the following ring, as also shown in Fig. 31.

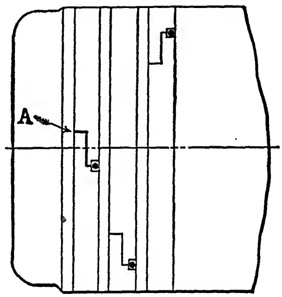


FIG. 31.

THE PISTON is made in one piece, the rings being sprung on over the junk-rings. It should be remembered that with oil engines greater heat is evolved in the cylinder than in steam engines. Consequently the slightest play is allowed to the piston-rings at the sides, and are, therefore, not made so tight a fit as in steam-engine practice.

THE CONNECTING-ROD BEARINGS at piston end are

scraped in the ordinary way, and should be allowed slight play sideways on the gudgeon-pin. In smaller-sized engines 1-64″ can be allowed, this amount being slightly increased in the larger-sized engines. The crank-pin bearing of the connecting-rod is usually allowed a very slight play sideways also.

THE AIR AND EXHAUST VALVES should not be a very close fit in their guides. If the fit in these guides is made too close when the valve-box becomes heated the consequent expansion may cause the valve-stem to stick in the guides, and leakage of the valve will result.

The valve-seats are by some considered best left sharp, being not more than 1-32″ wide before grinding.

THE WATER-JACKETS of cylinder or valve-boxes should be all tested by hydraulic pressure to at least 120 lbs. pressure per square inch before the piston is put into the cylinder.

THE FLY-WHEELS require careful keying onto crank-shaft. If the keys are not a good fit and not driven home tight the engine may knock when running. Two keys in larger-sized engines are usually supplied, one being a sunk key, which is fitted to keyway in recessed shaft as well as to the keyway cut in the fly-wheel hub, the second key being only recessed in the fly-wheel and being concave on the lower side to fit the shaft.

OIL-SUPPLY PIPES which have to withstand pressure should have the fittings "sweated" on, the unions being screwed into place on the brass or copper pipe while the solder is still in a liquid state.

CYLINDERS made of two or more parts require the joints of internal sleeve to be made with great care.

Asbestos or a copper ring is used to make this joint; sometimes wire gauze with asbestos is used, which has been found to give very good results.

CYLINDER LUBRICATORS.—The lubrication of the piston in explosive engines is of great importance. On those engines where it is convenient to use it, a mechanical type of lubricator is added. This device consists of an oil reservoir into which a wire attached to a revolving spindle is periodically dipped, the wire being also arranged to wipe over a projection which conducts the oil to a receptacle placed above the reservoir and connected to the top of the cylinder.

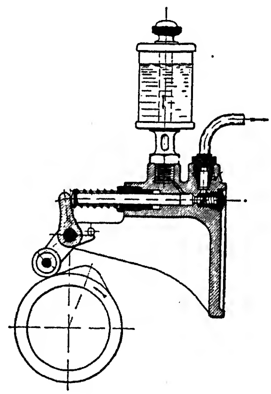


FIG. 31*a*.

The most efficient and economical lubricator for the piston is the force feed system shown in Fig. 31*a*, where the lubricant is forced by pump and reaches the piston at the proper time and position for best results in lubrication.

[Tables giving the Calorific Values of Oils, etc., will be found at end of Book.]

CHAPTER III.

TESTING ENGINES.

THE chief object in testing explosive engines at the factory is to ascertain that, in actual working at different loads, the several adjustments are correct. In the steam engine a physical process is completed, requiring only the inlet, expansion, and the outlet of the steam to and from the cylinder, whereas in the oil engine a chemical process is gone through consisting of the introduction of the proper mixture of vaporized oil and air into the cylinder, the ignition of this explosive mixture and the consequent combustion. All this must be accomplished before the piston receives an impulse. In order, therefore, that the best results be obtained, the different mechanisms controlling these processes are each set, and record of their performance during the test is taken with the indicator, which results are again verified by some form of brake attached to the fly-wheels or pulley of the engine, and are further checked in an oil engine by the record of the amount of oil which is consumed for the power developed. Where more detailed tests are required, the temperature of the exhaust gases, the amount of air consumed in the cylinder, its temperature and barometrical pres-

sure, together with the amount of cooling water necessary to keep the cylinder to the required temperature, are each noted and recorded. When the test is made with a new engine, it should be first started up and run without anv load for a short time. The cams are set as

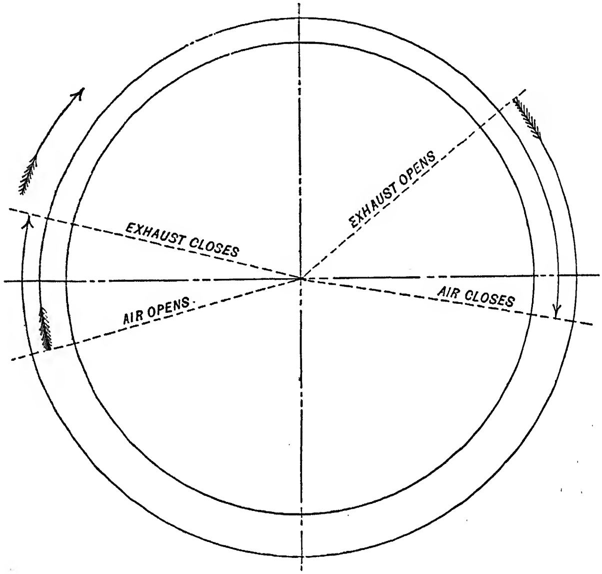


FIG. 32.

shown in diagram, Fig. 32, for engines having both air and exhaust valves actuated from the crank-shaft. The air-valve closes, as shown, just after the crank-pin has passed the out centre, the exhaust-valve opening at about 85 per cent. of the full stroke and closing just

after the air-valve has opened. Where the air-inlet valve is automatic the exhaust-cam only is set, as shown in the diagram, and the air-valve spring should be adjusted so that the incoming air is not choked in passing the valve during the suction stroke.

The oil-pipes leading to the vaporizer or sprayer should be well washed before starting the engine, as with a new engine grit and filings may get into the pipes, and when the engine is started the oil-valves and valve-seats may be damaged. The oil-filter also must be in proper shape and clean, so that the oil can flow freely to the oil-pipe.

After the vaporizer and igniter has been well heated a little oil should be allowed to enter the vaporizer or combustion chamber; then the fly-wheels can be turned forward a few times, after which the engine should start freely. The method of starting the different types of engines is explained in detail in Chapter VII. An engine is sometimes found difficult to start the first time owing to some defect in the castings or workmanship, and if it fails to start, the engine should be examined in detail to ascertain the cause.

First test the oil-inlet or spraying device by hand; then test the pressure of compression in the cylinder by turning the fly-wheels backward. The relief-cam being out of action, it should not be possible with full compression to turn the fly-wheel past the back centre. If the compression is so slight that the pressure in the cylinder can be overcome and the fly-wheel turned during the compression period by hand, then either the piston-rings are leaking or there is leakage past

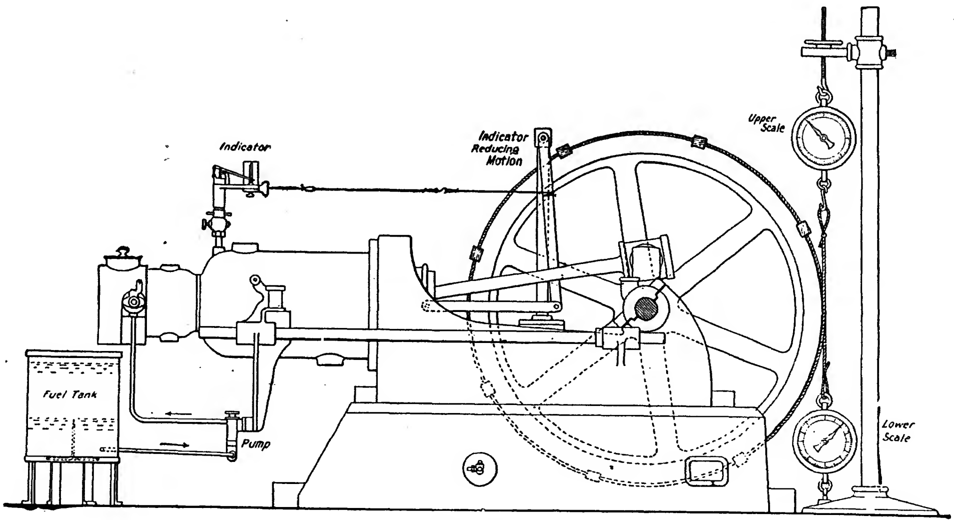

Indicator
Indicator Reducing Motion
Upper Scale
Lower Scale
Fuel Tank
Pump

FIG. 33.

the air and exhaust valves or through some of the joints or gaskets. Air and exhaust valves and piston-rings should be examined, and any appearance of leakage remedied by refitting the piston-rings, as already explained in Chapter II., and the valves, if necessary, should be reground in. New engines also fail to start at times by reason of the leakage of water from the cooling jacket into the cylinder owing to faulty gaskets or flaws in the castings. This leakage of water may sometimes be ascertained by failure to obtain an explosion in the combustion chamber when all conditions in the cylinder and vaporizer are apparently in good order for the engine to start properly. If leakage of water is suspected but cannot be detected in this way, the water-pressure pump should be attached and the water-jackets tested to a pressure of 120 lbs. The crank-shaft and other bearings require careful oiling at first, and full lubrication should be given to the piston; otherwise it may, perhaps, work dry and cut the cylinder.

After working a few hours, the piston should be withdrawn and examined; any hard places on the sides should be eased either by careful hand filing or otherwise. The junk-rings (or distance-pieces between the rings) should be eased if necessary, so that they do not work hard on the cylinder. The full bearing of the piston should be from about $\frac{1}{2}''$ from rings forward to within $\frac{3}{4}''$ of the front end, as already explained in Chapter II.

The terms "brake," or "developed," or "actual" or "effective" H. P., are synonymous, and are used

to signify the power which an engine is capable of delivering at the fly-wheel or belt-pulley. This power is variously designated, and here we shall use the abbreviation B. H. P. to express it. The indicated H. P. represents the whole power developed by combustion in the cylinder, but it is not considered such a reliable method of measuring the power of explosive engines as that of the dynamometer or brake, because the indicator-card only gives the power developed by one or more explosions, whereas the brake can be applied for any length of time and shows the average performance of the engine for a longer period of time.

Fig. 33 illustrates the engine as arranged for testing in the factory. The fuel tank shown at the left hand is placed there for the purpose of running the oil-consumption test. The fuel pump is connected temporarily to this tank instead of taking its supply of oil from the tank in the base of the engine. The indicator is also shown in place on the top of the cylinder. The device for reducing the stroke of the crank to suitable dimensions for the indicator is also shown in place bolted to the bed-plate of the engine. The brake consists of rope $\frac{1}{2}''$ thick, with wooden guides with balances at each extremity. The upper balance is suspended by adjustable hook suitably arranged for altering the load on the brake.

Various kinds of dynamometer brakes are used for testing; that shown in Fig. 33 is considered by the writer as being satisfactory. The brake should be attached as shown in the illustration, the load being taken as the number of pounds shown on the upper

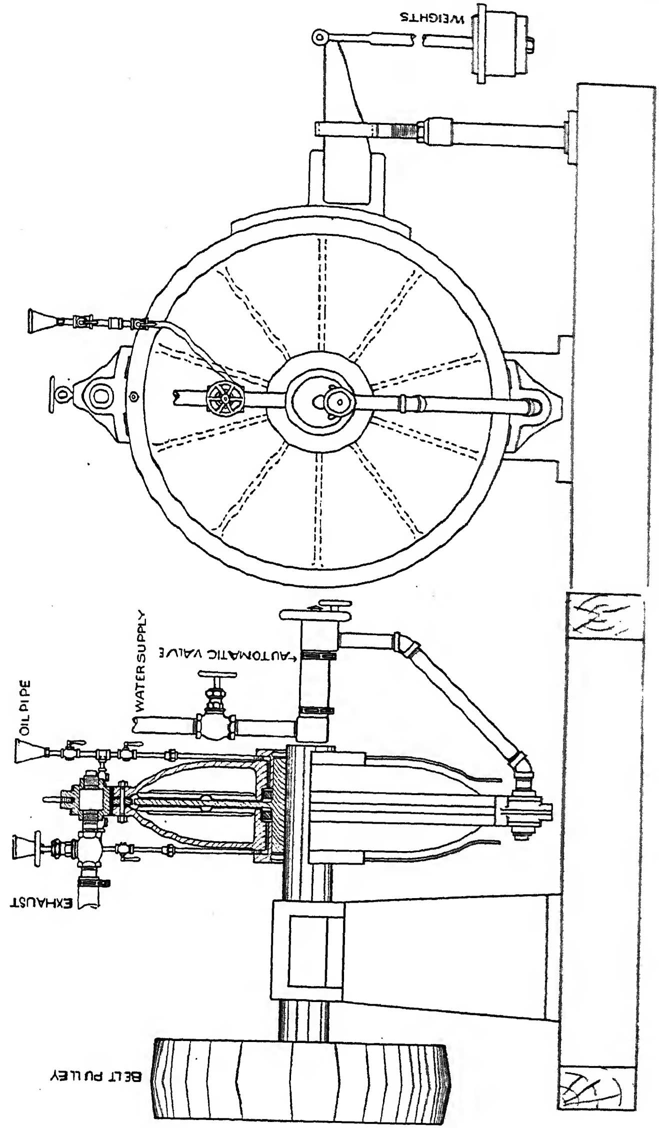

WEIGHTS
WATER SUPPLY
AUTOMATIC VALVE
OIL PIPE
EXHAUST
BELT PULLEY

FIG. 33a.

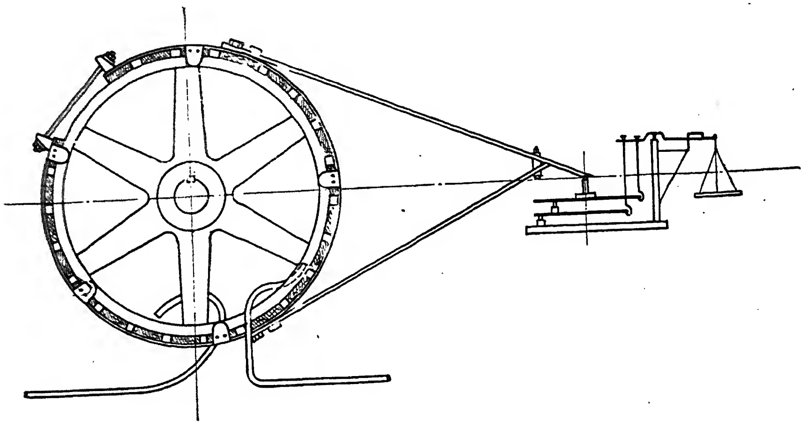


FIG. 33*b*.

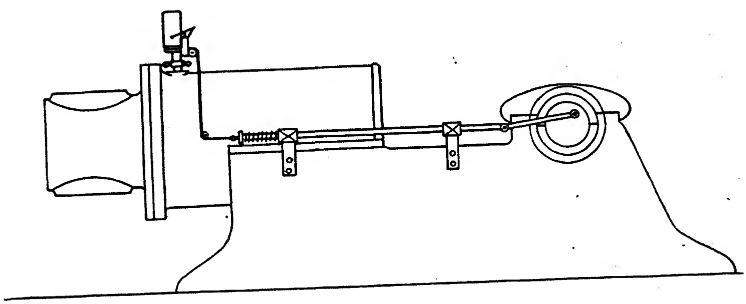


FIG. 34.

scale less those shown on the lower scale. Brake or actual H. P. is calculated thus:

$$\text{B. H. P.} = \frac{W \times C \times N}{33,000}.$$

W = net load in pounds.
C = circumference of wheel.
N = number of revolutions per minute.

The circumference of the wheel should be measured at the centre of the rope, thus allowing for half the rope thickness.

The Prony brake being water cooled is recommended for larger engines.

The power developed with this brake as shown in Fig. 33b is ascertained as follows:

$$\text{B. H. P.} = \frac{2R \times \pi \times l \times Q \times n.}{33.000}$$

When R = radius of wheel in feet.
Q = weight in pounds on scale + weight of brake apparatus.
l = distance in feet from center of shaft to point of contact of lever with scale.
π = 3.1416.
n = R. P. M.

The Alden dynamometer or absorption brake shown at Fig. 33a is advantageously used for measuring the horse-power when the prony brake or rope brake cannot be used. The power developed is calculated in the same way as with the prony brake, Fig. 33b. The dynamometer can be operated by belt or direct connected to the shaft of the engine.

THE INDICATOR is attached to the cylinder by first screwing into the cylinder the indicator cock, as shown at Fig. 34*a*, to which the indicator is applied in the ordinary way.

The length of the stroke of the engine must be reduced to suit the dimensions of the diagram, which is

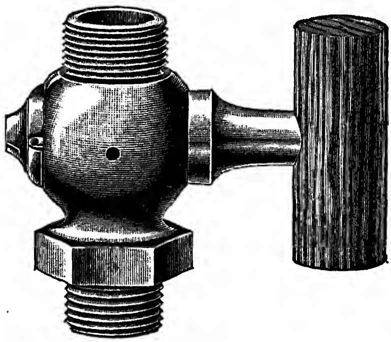


FIG. 34*a*.

usually about 3″ long. This is accomplished by the use of a device, as shown in Fig. 35 or 35*b*.

Indicated H. P. is calculated thus:

$$\text{I. H. P.} = \frac{P\,L\,A\,E}{33{,}000}.$$

P = mean effective pressure in lbs.
L = length of stroke in feet.
A = area in inches of piston.
E = number of explosions per minute.

The M. E. P. of indicator-card is obtained by the use of the planimeter, as shown in Fig. 37, or by measuring the card by scale and taking the average pressure.

The illustration (Fig. 36) shows the design and

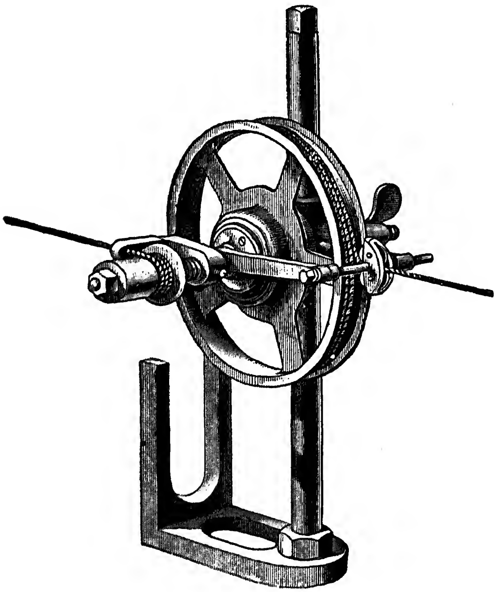


FIG. 35.

arrangement of the parts of the Crosby gas-engine indicator. The cylinder proper is that in which the movement of the piston takes place. The piston is formed from a solid piece of tool steel, and is hardened to prevent any reduction of its area by wearing. Shal-

low channels in its outer surface provide an air packing, and the moisture and oil which they retain act as lubricants, and prevent undue leakage by the piston.

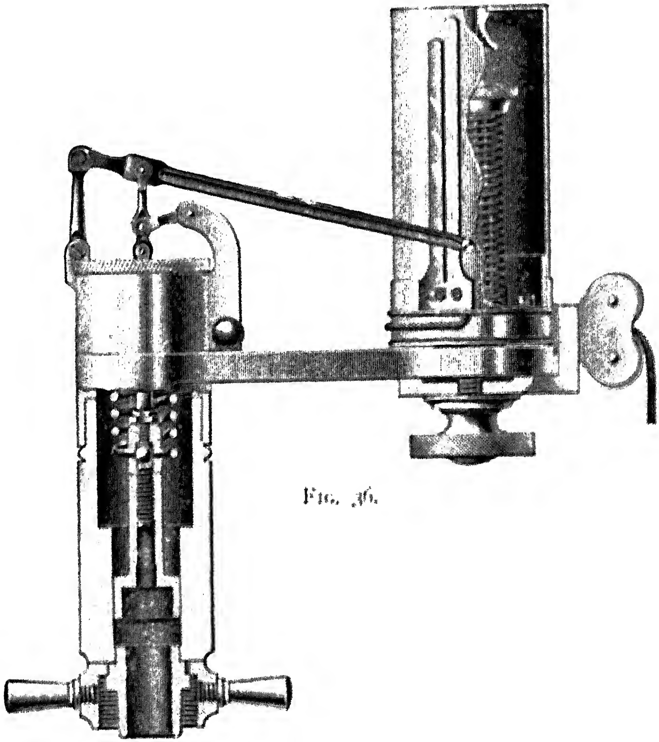


FIG. 36.

The piston is threaded inside to receive the lower end of the piston-rod and has a longitudinal slot which permits the bottom part of the spring with

its head to drop on to a concave bearing in the upper end of the piston-screw, which is closely threaded into the lower part of the socket; the head of this screw is hexagonal, and may be turned with a hollow wrench.

The swivel-head is threaded on its lower half to screw into the piston-rod more or less according to the required height of the atmospheric line on the diagram. Its head is pivoted to the piston-rod link of the pencil mechanism. The pencil mechanism is designed to eliminate as far as possible the effect of momentum, which is especially troublesome in high-speed work. The movement of the spring throughout its range bears a constant ratio to the force applied, and the amount of this movement is multiplied six times at the pencil point.

SPRINGS.—In order to obtain a correct diagram, the height of the pencil of the indicator must exactly represent in pounds per square inch the pressure on the piston of the oil engine at every point of the stroke; and the velocity of the surface of the drum must bear at every instant a constant ratio to the velocity of the engine piston.

THE PISTON SPRING is made of a single piece of spring steel wire, wound from the middle into a double coil, the spiral ends of which are screwed into a brass head having four radial wings to hold them securely in place; 80 to 200 lb. spring is a suitable pressure for this work.

This type of indicator is ordinarily made with a drum one and one half inches in diameter, this being

the correct size for high-speed work, and answering equally well for low speeds.

To remove the piston and spring, unscrew the cap; then take hold of the sleeve and lift all the connected parts free from the cylinder. This gives access to all the parts to clean and oil them.

To change the location of the atmospheric line of the diagram.—First, unscrew the cap and lift the sleeve, with its connections, from the cylinder; then turn the piston and connected parts toward the left, and the pencil point will be raised, or to the right and it will be lowered. One complete revolution of the piston will raise or lower the pencil point $\frac{1}{8}$", and this should be the guide for whatever amount of elevation or depression of the atmospheric line is needed.

To change to a left-hand instrument.—If it is desired to make this change: First, remove the drum, and then with the hollow wrench remove the hexagonal stop screw in the drum base, and screw it into the vacant hole marked *L*; next, *reverse* the position of the adjusting handle in the arm; also, the position of the metallic point in the pencil lever; then replace the drum, and the change from right to left will be completed.

The tension on the drum spring may be increased or diminished according to the speed of the engine on which the instrument is to be used, as follows: Remove the drum by a straight upward pull; then raise the *head* of the spring *above* the *square* part of the spindle, and turn it to the right for more or to the left for less tension, as required; then replace the head on the spindle.

Before attaching the indicator to an engine, allow air to blow freely through pipes and cock to remove any particles of dust or grit that may have lodged in them.

The indicator should be attached close to the cylinder whenever practicable, especially on high-speed engines. If pipes must be used they should not be smaller than half an inch in diameter, and as short and direct as possible.

The indicator can be used in a horizontal position, but it is more convenient to take diagrams when it is in a vertical position, and this can generally be obtained, when attaching to a vertical engine, by using a short pipe with a quarter upward bend.

The motion of the paper drum may be derived from any part of the engine, which has a movement coincident with that of the piston. In general practice and in a large majority of cases the piston itself is chosen as being the most reliable and convenient.

When the indicator is in position and the cord-drum or other reducing motion is correctly placed, it is next necessary to adjust the length of the cord, so that the drum will clear the stops at each extreme of its motion. The engine should be allowed to run for a few minutes to heat up before taking a diagram. The atmospheric line should be drawn by hand, preferably after the diagram has been taken and when the instrument is heated up; the card is then taken with full-rated load on the brake. It is well to allow the pencil to go several times over the paper so as to procure a card showing several explosions, and thus the average pressure can be taken.

The pressure of the pencil on the paper can be adjusted by screwing the handle in or out, so that when it strikes the stop there will be just enough pressure on the pencil to give a distinct fine line. The line should

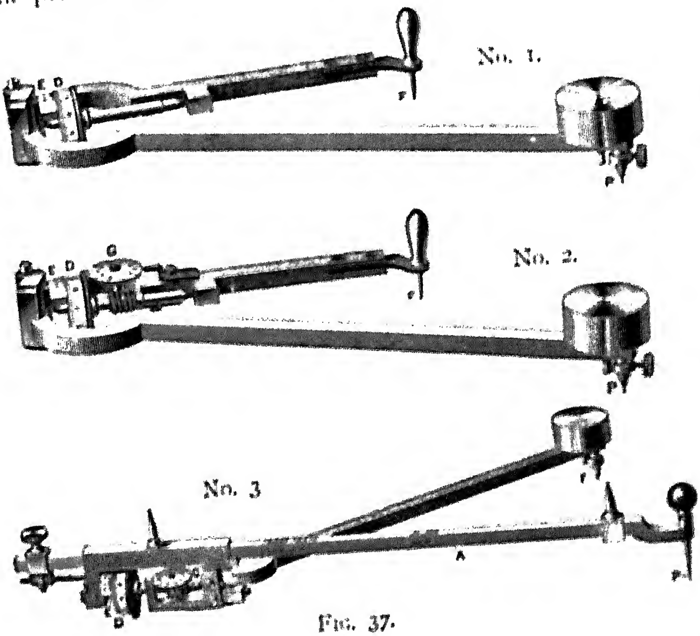


FIG. 37.

not be heavy, as the friction necessary to draw such a line is sufficient to cause errors in the diagram.

THE PLANIMETER or averaging instrument is shown at Fig. 37. No. 1 planimeter is the simplest form of the instrument, having but one wheel, and is designed to measure areas in square inches and decimals of a

square inch. The figures on the roller wheel *D* represent *units*, the graduations *tenths*, and the vernier *E* gives the *hundredths*. *F* is the tracer and *P* is the pivot.

Fig. 37 represents the No. 2 planimeter, which is the same as the No. 1, with the addition of a counting disc *G*, the figures on which represent *tens* and mark complete revolutions of the roller-wheel. By this means areas greater than ten square inches can be measured with facility. The result is given in square inches and decimals, and the reading from the roller wheel and vernier is the same as with No. 1.

Fig. 37 represents the No. 3 planimeter, which differs somewhat in design from the two previously described. It is capable of measuring larger areas, and by means of the adjustable arm *A* giving the results in various denominations of value, such as square decimeters, square feet and square inches; also of giving the average height of an indicator diagram in fortieths of an inch, which makes it a very useful instrument in connection with indicator work.

Directions for Measuring an Indicator Diagram with a No. 1 or No. 2 Planimeter.

Care should be taken to have a flat, even, unglazed surface for the roller wheel to travel upon. A sheet of dull-finished cardboard serves the purpose very well. Set the weight in position on the pivot end of the bar *P*, and after placing the instrument and the diagram

in about the position shown in Fig. 37*a*, press down the needle point so that it will hold its place, set the tracer; then at any given point in the outline of the diagram, as at *F*, adjust the roller wheel to zero. Now follow the outline of the diagram carefully with the tracer

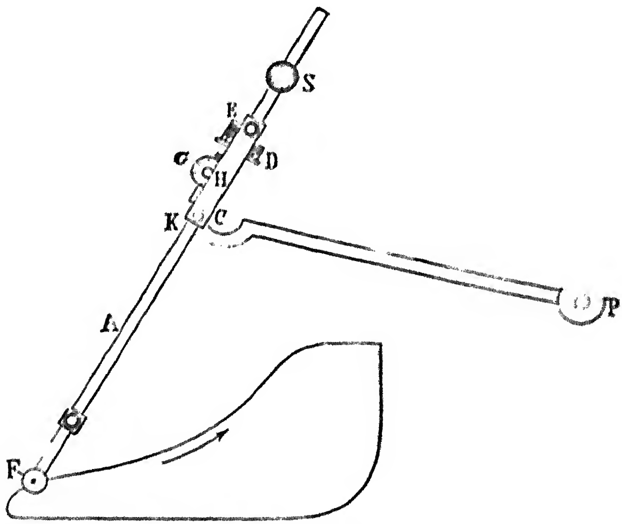


FIG. 37*a*.

point, moving it in the direction indicated by the arrow, or that of the hands of a watch, until it returns to the point of beginning. The result may then be read as follows: Suppose we find that the largest figure on the roller wheel *D* that has passed by zero on the vernier *E* to be 2 (units) and the number of graduations that have also passed zero on the vernier to be 4

(tenths), and the number of graduation on the vernier which exactly coincides with the graduation on the wheel to be 8 (hundredths), then we have 2.48 square inches as the area of the diagram. Divide this by the length of the diagram, which we will call 3 inches, and we have .8266 inch as the average height of the diagram. Multiply this by the scale of the spring used in taking the diagram, which in this case is 40, and we have 33.06 pounds as the mean effective pressure per square inch on the piston of the engine.

Directions for Using the No. 3 Planimeter.

No. 3 planimeter is somewhat differently manipulated, although the same general principle obtains. The figures on the wheels may represent different quantities and values, according to the particular adjustment of the sliding arm *A*. If it is desired merely to find the area in square inches of an indicator diagram, set the sliding arm so that the 10-square-inch mark will exactly coincide with the vertical mark on the inner end of the sleeve *H* at *K*. The sliding arm is released or made fast by means of the set-screw *S*.

With the wheels at zero and the planimeter and diagram in the proper position, trace the outline carefully and read the result from the roller wheel and vernier, the same as directed for the No. 1 and No. 2 instruments.

The indicator-card shows what is occurring inside the cylinder and combustion chamber during the different periods of the revolution. It gives a record of the

variations in pressure, and also the exact points of the opening and closing of the valves. With the Otto or Beau de Rochas cycle the four strokes are as follows: Suction (*A*), compression (*B*), expansion (*C*), exhaust (*D*). The lines in the diagram are correspondingly lettered (see Fig. 38), and they represent each of these processes.

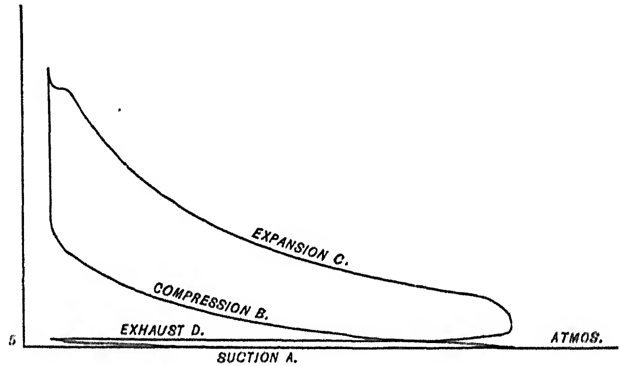


FIG. 38.

Fig. 39 shows a good working diagram, in which the mixture of air and hydrocarbon gas is correct and where combustion is practically complete. The ignition line in this diagram is nearly perpendicular to the atmospheric line, but inclines slightly toward the right hand at top. The diagram also shows the opening of the exhaust-valve at the proper time—namely, at 85 per cent. of the stroke. The compression line represents the proper pressure, and the air-inlet and exhaust lines indicate correct proportioned valves and inlet and outlet passages.

In considering and analyzing diagrams the following hints will perhaps be of service. If the suction line of the diagram is shown below the atmospheric

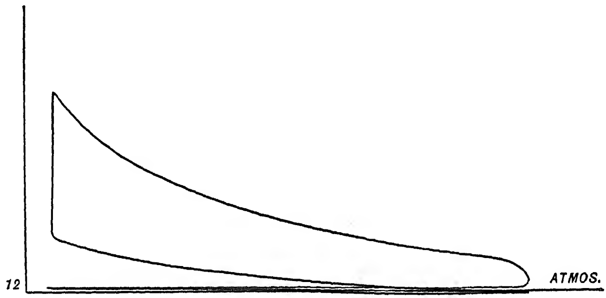


FIG. 39.

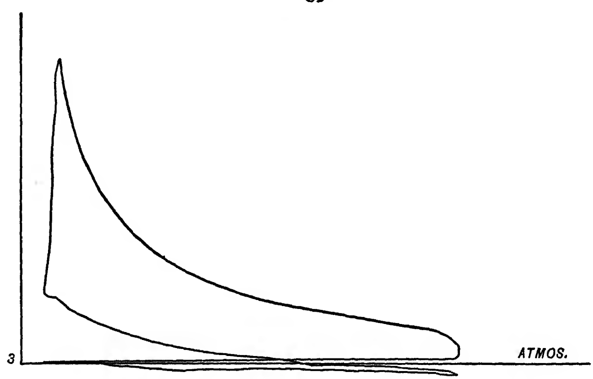


FIG. 40.

line, as in Fig. 40, then the air-inlet to the cylinder is known to be in some way choked. Where the air-valve is automatic this defect may be caused by the valve-

spring being too strong and it accordingly requires weakening; or the area of the air suction-pipe, if this is used, may be too small or this connection may have too many elbows or bends in it, and should be either of increased diameter or the bends should be eliminated. Again, the valve itself may have too small an area, or if actuated have insufficient lift (the proper lift of a valve is $\frac{1}{4}$ of its diameter), or the period of opening of the valve may not be correct, and the setting of the cams should be carefully examined, and, if necessary, altered in accordance with the diagram of valve opening, as shown at Fig. 32.

If the compression line *B* shows insufficient pressure of compression, this indicates leakage, which is probably due either to leaky piston or valves. If this leakage is past the piston-rings, the escaping air may be heard and the lubricating oil will be seen at each explosion period to be splashing and blown past the rings of the piston. If no signs of piston leakage are noticed, then examine oil-inlet air and exhaust valves and valve-seats very carefully; also note the various joints in the valve-box and otherwise where leakage might possibly occur. In engines without water-jackets around the valve-box the heat of the exhaust gases continually passing through the valve-chamber may sometimes cause the valve-seats to expand unequally when heated, and consequent leakage will occur when working.

If leakage is detected at the valves they must be reground, and also any hard places on the valve-stems or guides where they become heated should be eased so that the valves will work easily and efficiently when the

seats and guides are expanded, and, perhaps, slightly distorted, by the heat of working. (It is understood that these remarks refer to new engines solely.) With some engines means of increasing the compression by movable plates on the connecting-rod crank-pin end or other somewhat similar means are provided which can be changed, if necessary, thus decreasing the

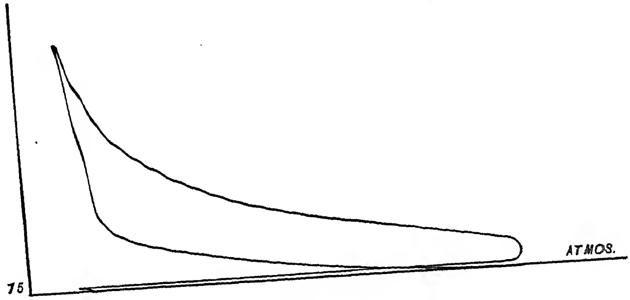


FIG. 41.

amount of clearance in the cylinder. If the piston-rings are without leakage and they have worked into their proper bearings in the cylinder, and if all the valves are in perfect order and without leakage, and still the compression pressure, as shown on the diagram and as already explained, requires increasing, then the clearance in the cylinder can be slightly decreased where it is possible to do so. The vertical ignition line shows the timing of the ignition, and also the initial pressure of explosion. If this line is as represented in Fig. 41 the ignition is known to be too early, and should be arranged to occur somewhat later. The

diagrams as shown in Fig. 42 has the ignition line too late.

The timing of the ignition is regulated as follows:

With electric ignition by altering the period of

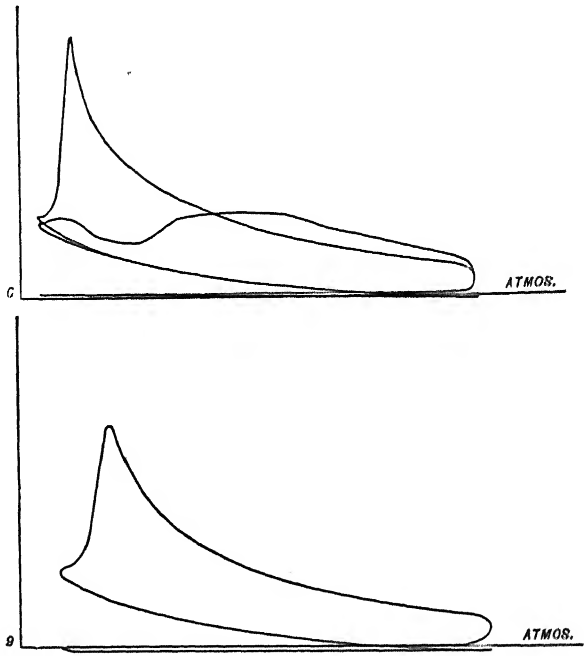


FIG. 42.

sparking. Thus, if later ignition is required the igniting device must not be allowed to spark till the crank-pin has travelled nearer to the dead centre. With the hot-tube ignition and no timing valve, the length of the

tube can be changed. For example, to retard the ignition the tube should be lengthened slightly and its temperature somewhat decreased. In engines where neither of these means of ignition is used, but where the ignition is caused by the heat of the vaporizer-chamber or somewhat similar device, the timing of the ignition is controlled by the heat of the vaporizer-chamber and also by the heat generated by the process of compression. Where the ignition in this case is to be retarded, the compression should be reduced slightly and the vaporizer or other igniting device maintained at a less heat. The ignition, however actually caused, is always influenced by the heat of the cylinder walls and the temperature of the incoming air, which correspondingly increases or decreases the heat caused by the compression before explosion takes place. The ignition is usually adjusted when testing engines with the cooling water issuing from the cylinder water-jackets at a temperature of 110° to 130° Fahr.

The expansion line is marked *C*, as shown in Fig. 38. This line indicates the initial pressure of combustion, and it also shows the developed pressure decreasing as the volume of the cylinder becomes greater with the piston moving forward. The effective pressure developed is measured from this line to the compression line, and varies according to the richness of the explosive mixture. When the engine is in actual use the governor controls this pressure automatically.

The mean effective pressure is greater in some types of engines than it is in others, and varies, as stated in Chapter II., from 40 to 75 lbs. The amount of the

pressure in the cylinder is dependent upon the method of vaporization, upon the proper mixture of the gas

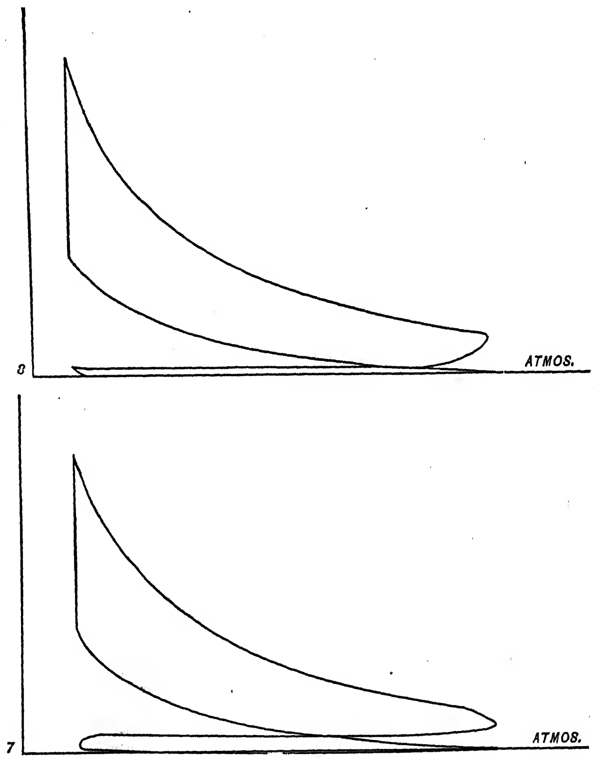


FIG. 43.

and air before explosion, and also upon the press of the compression. As in gas engines, the tendenc oil-engine practice is toward higher compression

increase their efficiency. Where the mean effective pressure is low the relative power of the engine will, of course, also be reduced. The greatest mean effective pressure should be attained when the oil is thoroughly vaporized, is properly mixed with the air and when the compression is as high as practicable without pre-ignition taking place.

Should the exhaust lines *D* appear as in Fig. 43, then it is understood that the discharge of the exhaust gases is in some way choked; this may be caused by the exhaust-valve itself being too small, or to the periods of the opening of the valve being incorrect. (See diagram, Fig. 32.) Again, this defect may be caused by too many sharp bends, too small diameter exhaust-pipe, or possibly too long an exhaust-pipe. Theoretically no back pressure should be allowed during the exhaust period, but usually in practice a slight pressure of about one pound is recorded.

Each pound per square inch of back pressure shown by the exhaust line shows a back pressure in the cylinder, which is negative work to be overcome by the piston, and represents a slight loss of power by the engine.

Care must be taken that the indicator is in proper condition, without any play in the pencil arm, and that the piston is free and well lubricated. Lost motion in the indicator may show peculiarities in the diagram which to an inexperienced manipulator may be the cause of trouble.

TACHOMETERS (Fig. 44).—These instruments have been designed for the purpose of ascertaining at a

glance the number of revolutions made in a given time by rotating shafts. Their construction is based on centrifugal power, and they consist of a case inside of which are mounted a pendulum ring, in connection with a fixed shaft, a sliding rod and an indicating

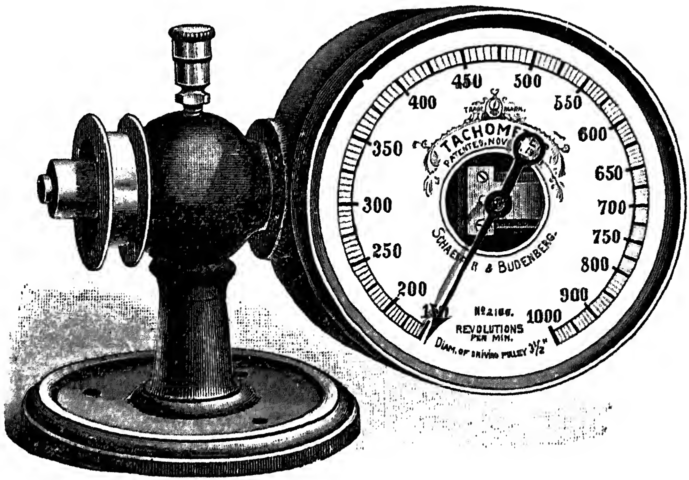


FIG. 44.

movement. The apparatus is very sensitive, and will indicate the slightest deviation in speed.

PORTABLE TACHOMETER (Fig. 44*a*).—This instrument is similar in construction to the tachometer for permanent attachment. By applying it by hand to the centre of rotating shafts, it will instantly and correctly indicate the number of revolutions of the shaft per minute.

Fig. 44*b* illustrates a new form of speed counter, the

invention of Mr. A. J. Hill, of Detroit, Mich., which, besides counting, also registers the number of revolu-

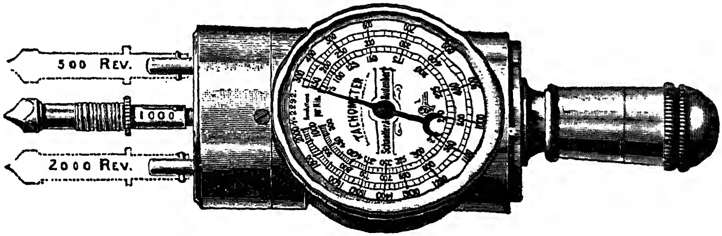
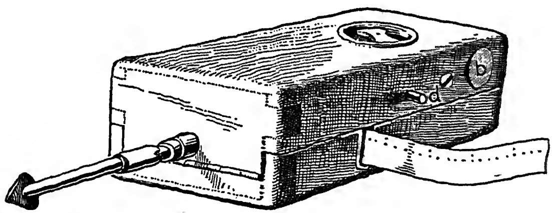


FIG. 44*a*.

tions of the shaft. This is accomplished by simply punching a continuous slip of paper, as shown in



44*b*.

Fig. 44*c*. The watch mechanism in the device also periodically records a detent in the paper slip, thus

44*c*.

marking the periods of time while the shaft actuates the mechanism of the device, causing a detent for each

revolution. The writer has not yet had an opportunity of testing this interesting and useful invention.

When the full brake H. P. is obtained, which should be developed for at least a period of one hour continuously, the consumption fuel test is made.

THE MECHANICAL EFFICIENCY of oil engines, as shown by records of various tests, should be from 80 per cent. to 88 per cent., although the efficiency is much less than this when the engine has been working only a short time and before the crank-shaft and other bearings and piston are worn in. To ascertain the mechanical efficiency of an engine, first calculate the I. H. P., as already described; then figure the B. H. P., as already shown. Then:

$$\text{Mechanical efficiency} = \frac{\text{B. H. P.}}{\text{I. H. P.}}$$

For instance: If the B. H. P. of an engine = 10 and the I. H. P. = 12.5,

$$\text{Mechanical efficiency} = \frac{10}{12.5}$$
$$= 80 \text{ per cent.}$$

THERMAL EFFICIENCY.—The ratio of the heat utilized by the engine, as shown by the power (B. H. P.) developed, as compared with the total heat contained in the fuel absorbed by the engine, is known as the thermal efficiency. This can be obtained by the following formula:

$$\frac{42.63 \times 60}{C \times X}.$$

C = consumption of fuel in pounds per B. H. P. per hour.

X = calorific value of the fuel per pound in heat units.

The thermal efficiency of different makes of oil engines varies. In the older type of engines a thermal efficiency of 15 per cent. was the maximum, as shown by the following disposition of heat by Mr. Dugeld Clerk, applicable to older engines. In the modern engines (see test, page 248) a thermal efficiency equivalent to approximately 28 per cent. has been obtained.

Heat shown on diagrams per I. H. P. .	15.3 per cent.
Heat rejected in water-jackets.......	26.8 per cent.
Heat rejected in exhaust and other losses.........................	57.9 per cent.
	100 per cent.

The above table of disposition of heat is applicable to smaller engines. The efficiency of the gas engine is approximately 27 per cent., while the efficiency of the complete steam plant does not exceed 12 per cent.

FUEL CONSUMPTION TEST.—This is generally made with all new engines before they leave the factory, and is advantageous as a check of the efficiency of the engine as shown by the indicator and the brake tests, and this test is also useful to ascertain the exact consumption of fuel by the engine in actual operation.

The oil is weighed, the amount being gauged by weight of fuel rather than by measuring the oil. The tank or other receptacle from which the fuel is drawn is first filled with kerosene. The tank is then placed on platform scales, and the weight is carefully taken and time noted when the engine is ready to begin this test. The full load required is then adjusted on the brake while the engine is running at its normal speed.

The oil can also be measured by means of a pointer placed in the tank, the tank being filled until the pointer is just visible before the engine is ready for the test to commence. The oil is then weighed in a separate vessel, and a quantity of the fuel is poured into the test tank. When the test is completed, the oil is taken out of the tank until the pointer shows again just as it did at the commencement of the test. The weight of the kerosene remaining in the vessel is deducted from the whole weight as at first recorded, and the difference is the amount consumed by the engine. It is usual to continue this test for at least one hour's duration. During the consumption test, the load on the brake and the number of revolutions per minute are recorded and the average brake horse-power developed is taken. The exact amount of oil consumed per hour being also known, the consumption of oil per H. P. hour is simply ascertained.

Light spring indicator diagrams are taken to ascertain the efficiency of the air and exhaust valves, ports and passages. That shown at Fig. 45 is taken with $\frac{1}{20}$ spring. The indicator must be fitted with special stop arrangement to prevent the pencil going above

the drum of the indicator when taking light spring cards.

It is advantageous to have some method of limiting the supply of oil to the vaporizer arranged so as to prevent the engine from consuming an excess of oil at any time. This gauge should be made immediately after the consumption test has been proved as satisfactory, and to avoid possible mistake by alteration of the oil supply. As already described, if too much oil enters

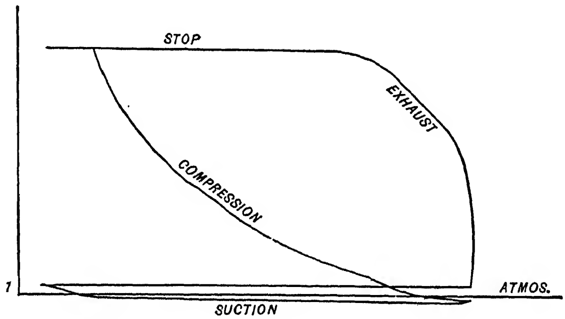


FIG. 45.

the vaporizer, bad combustion will follow and carbonization will, perhaps, result, thus rendering the piston sticky and gummy, and materially reducing the efficiency of the engine.

The exact periods for the movements of the valve and cams should also be clearly marked on the gearing or elsewhere, so that if at any future time the crank-shaft is taken out or the gearing (or other mechanism) between the crank-shaft and the cam-shaft removed,

the relative position of the crank-shaft with the valve mechanism can be readily ascertained and the exact position of the cams again found without difficulty.

EXHAUST GASES.—With an oil engine it is important to note the color of the exhaust gases, which may vary a little according to the weather. Where complete combustion is taking place, the exhaust gases are almost, if not entirely, invisible. When the engine is first started, these gases will, perhaps, be white, gradually getting bluer.

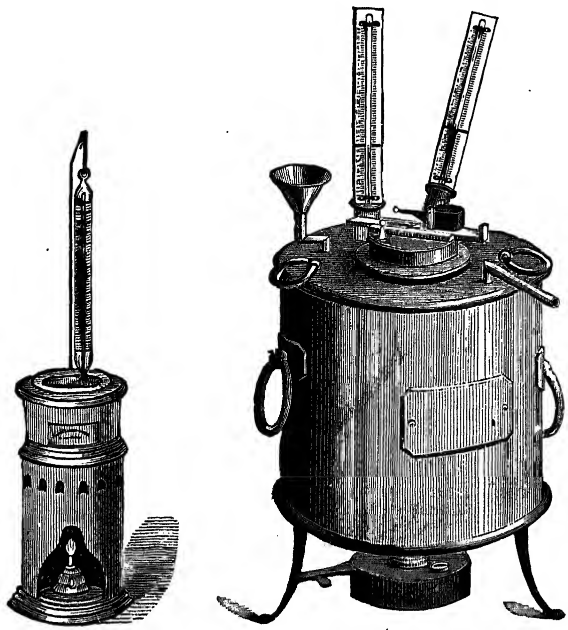
If an oil engine is working well and if the combustion is complete, the exhaust gases will not be seen but only heard, and the piston will also remain clean in working.

TESTING THE FLASH POINT OF KEROSENE.—Fig. 46*a* shows apparatus for ascertaining the "open fire" test or the temperature at which kerosene will flash or explode. This device consists of a small copper vessel in which the kerosene is placed. This vessel is immersed in a larger vessel containing water, which forms part of the upper part of the apparatus.

A thermometer is suspended with its lower part in the oil. A heating lamp placed under the receptacle containing the water raises the temperature of both water and oil as required. A lighted taper is passed to and fro over the top of the oil as it becomes heated. When the vapor given off by the oil flashes the temperature is noted, and that is termed the "flashing point" of the oil thus tested.

The "Abel" oil-tester is shown at Fig. 46*b*. This

was originated by Sir Frederick Abel, and hence its name. The tests made with this apparatus are those known as the "Abel closed" test. Such tests are recognized by the law (at the present time) of Great Britain.



a. *b.*

FIG. 46.

The device consists of a copper vessel containing water in which is an air-chamber. In the air-chamber is placed an oil-cup made of gun-metal. This oil-cup is supplied with tight-fitting lid, and is provided with gas

or oil lamp suitably arranged to ignite the oil vapor when required.

Two thermometers are required, one immersed in the oil and the other in the water, each having a tight joint around it.

The following are the instructions for performing this test: The heating vessel or water-bath is filled until the water flows out at the spout of the vessel. The temperature of the water at the commencement of the test is 130° Fahrenheit. The water having been raised to the proper temperature, the oil to be tested is poured into the petroleum cup, until the level of the liquid just reaches the point of the gauge which is fixed in the cup. If necessary, the samples to be tested should be cooled down to about 60°. The lid of the cup with the slide closed is then put on, and the oil-cup is placed in the water-bath or heating vessel, the thermometer in the lid of the cup being adjusted so as to have its bulb immersed in the liquid. The test-lamp is then placed in position upon the lid of the cup, the lead line, or pendulum, which has been fixed in a convenient position in front of the operator, is set in motion, and the rise of the thermometer in the petroleum cup is watched. When the temperature has reached about 66° the operation of testing is to be commenced, the test flame being applied at once for every rise of 1° in the following manner:

The slide is slowly drawn open while the pendulum performs three oscillations, and is closed during the fourth oscillation. Thus a flame is made to come in contact with the vapor above the oil. The temperature

at which the vapor flashes is noted, and is called the flashing point of the oil. If it is desired to employ the test apparatus to determine the flashing points of oils of very low volatility, the mode of proceeding is modified as follows:

The air-chamber which surrounds the cup is filled with cold water, to a depth of $1\frac{1}{2}$ inches, and the heating vessel or water-bath is filled with cold water. The lamp is then placed under the apparatus and kept there during the entire operation. If a very heavy oil is being dealt with, the operation commences with water previously heated to 120° instead of with cold water.

VISCOSITY OF OIL.—It is frequently advantageous to ascertain the viscosity of different oils. The device shown at Fig. 46*c* is manufactured by C. I. Tagliabue especially for this purpose. The viscosity of an oil with this apparatus is found by noticing the number of seconds required for fifty cubic centimetres of oil to pass the open faucet or valve.

To test the viscosity of oil at 212° Fahr. with this apparatus, first pour water into the boiler through opening *A*, unscrew safety-valve until water-gauge shows that the boiler is full, open stop-cock *B*, making a direct connection between the boiler and upper vessel which surrounds the receptacle in which the oil to be tested is placed. Suspend a thermometer so that its bulb will be about $\frac{1}{4}$ inch from the bottom of the oil-bath. After carefully straining 70 cubic centimetres of the oil to be tested, which must be warmed in the case of very heavy oils, pour same into the oil-bath. Close

stop-cocks *D* and *E*. Screw the extension *F* with rubber hose attached into the coupling *G*, and let the open end of the hose be immersed in a vessel of water,

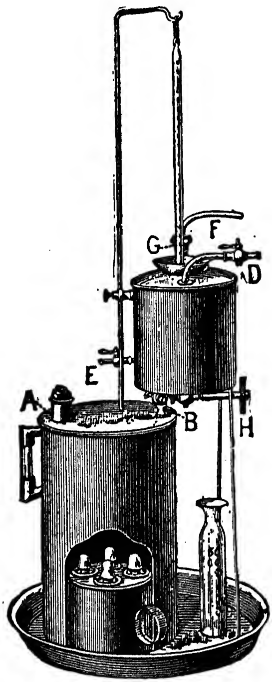


FIG. 46*c*.

which will prevent too large a loss of steam. Place lamp or Bunsen burner under boiler; screw steel nipple marked 212° on to stop-cock *H*; the apparatus is then ready to use. After steam is generated, wait until the

thermometer in oil-bath shows a temperature of from 209° to 211°; then place the 50 cubic centimetre glass under stop-cock *H*, so that the stream of oil strikes the side of test-glass, thereby preventing the forming of air-bubbles; and when the thermometer indicates its highest point open the faucet *H* simultaneously with the starting of the timing watch. When the running oil reaches the 50 cubic centimetre mark in the neck of the test-glass the watch is instantly stopped and the number of seconds noted.

To ascertain the viscosity, multiply the number of seconds by two, and the result will be the viscosity of the oil. For example: If 50 cubic centimetres of oil runs through in 101½ seconds, the viscosity will then be 203.

To test the viscosity of oils at 70° Fahr. screw the steel nipple marked 70 on to faucet *H*; close stop-cock *B*, closing communication between boiler and upper vessel; also close stop-cock *E*. Fill upper vessel through opening *G* with water at a temperature as near 70° as possible, also having the oil to be tested at the same temperature; hang the thermometer in position, and after stirring the oil thoroughly, blow through rubber tube at *D* to thoroughly mix the water; should the thermometer show higher or lower than 70° add cold or warm water until the desired temperature is attained. Then proceed as before stated.

[For tables of tests of various oil engines, see end of book.]

CHAPTER IV.

COOLING WATER-TANKS, AND OTHER DETAILS.

WATER is always required to keep the cylinders of explosive engines cool, and is necessitated by the great heat evolved in such engines, which heat would, if it were not carried away, prevent the proper working of an engine by too great expansion of the piston and by burning the lubricating oil. Where running water is not available, water-tanks are sometimes used. The engine water-jackets are connected to the tanks as shown in Fig. 47. It is important that the water piping rises all the way from the engine to the tanks. The water, when tanks are used, circulates by gravitation—that is, the cold water being slightly heavier than the hot sinks to the bottom of the tank, passes from the tank to the water-jacket, and returns as warm water to the top of the tank to be cooled off and again sink to the bottom of the tank.

The cooling water-tanks must be of not less capacity than 70 gallons of water per brake H. P. of engine. The tanks when installed should preferably be placed in the best location for cold air to circulate around

them, so that the water in the tanks may cool off as quickly as possible.

Where an engine is required to work for more than ten hours per day, the tanks should be of larger capacity than that above stated, or provision should be made

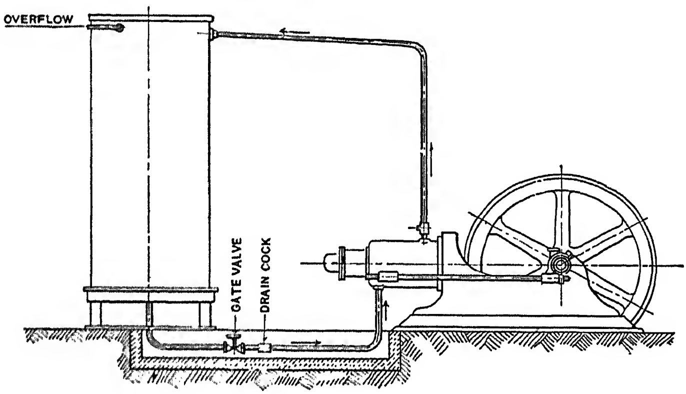


FIG. 47.

to add cold water to the tanks when the water becomes heated above 120° Fahrenheit.

The waste-water drain-pipe from the tanks should be arranged to allow the hot water to run off from the top of the tanks and the cold-water inlet-pipe arranged to enter near the bottom. The circulating-water pipes connecting the tanks to engine water-jacket should be large enough to allow the water to circulate freely. A pipe having $1\frac{1}{2}''$ inside diameter is considered suit-

able for the smaller size of engines and 3″ diameter pipe is sufficient for engines of 25 B. H. P. and over.

In some installations cooling water is available, but may require pumping to the engine. In such cases a pump capable of delivering more than ten gallons per brake H. P. of engine should be used. This pump can be actuated from the cam-shaft of engine as shown in Fig. 50, or from the crank-shaft by eccentric in the usual way. A rotary pump is sometimes used to accelerate the circulation of water in hot climates with the tank system of cooling water, and can be driven by belting from the crank-shaft of the engine. A by-pass in the water-pipes between the suction-pipe and the discharge-pipe of the water-circulating pump is advantageous, having a regulating valve in the by-pass. If this by-pass is not made, other means should be arranged, so that the supply of cooling water can be regulated to maintain the proper temperature of the cylinder of the engine—namely, 110° to 130° Fahrenheit. This temperature is recommended by the makers of several oil engines.

Where neither pump to lift and circulate cooling water nor water-tanks are necessary and where water is used from the city water-mains, ¾″ inside diameter pipe is sufficient for small and moderate-sized engines. The larger size may have 1″ diameter pipe connections to cylinder.

In all cases, either with tanks, water-pumps, or where the water is connected direct from the city water-main, provision must be made for emptying the cylinder water-jacket and all the water-pipes in time of

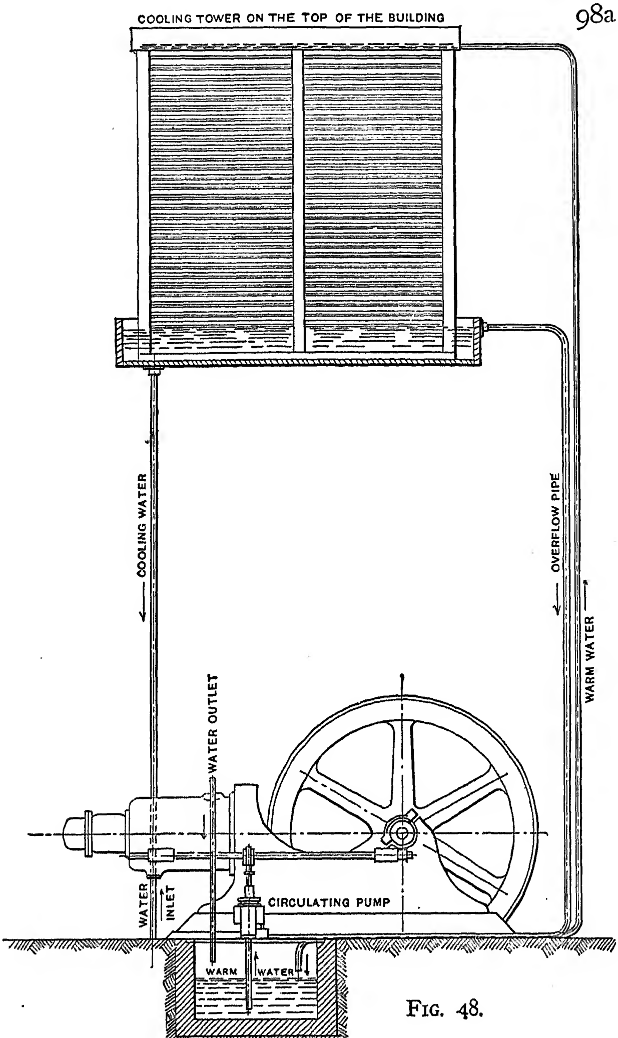

COOLING TOWER ON THE TOP OF THE BUILDING
COOLING WATER
OVERFLOW PIPE
WARM WATER
WATER OUTLET
WATER INLET
CIRCULATING PUMP
WARM WATER

FIG. 48.

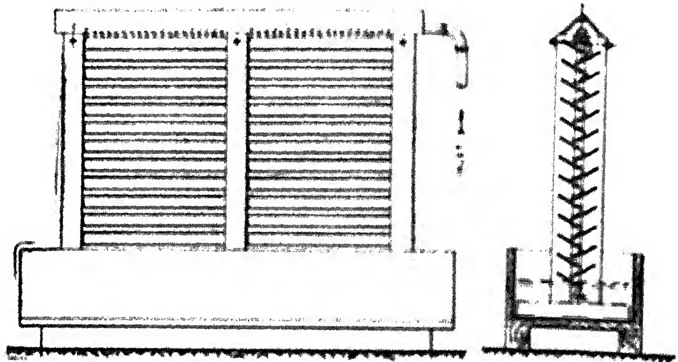


FIG. 48a.

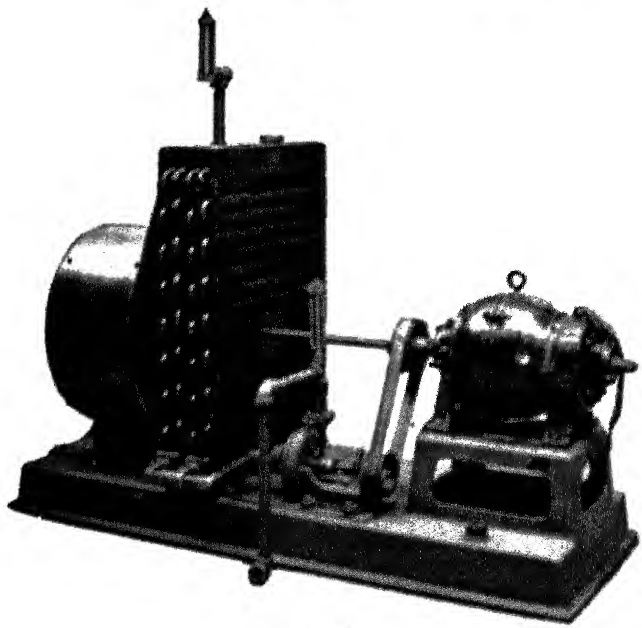


FIG. 48b.

frost. If the water in the water-jacket of the cylinder should be allowed to freeze, the cylinder casting may be cracked, and this may necessitate very expensive repairs.

RADIATORS FOR COOLING PURPOSES.—This is an apparatus for cooling the cylinder water of engines, sometimes used where space is not available for cooling tanks, and where the cooling tower shown in Fig. 48*b* cannot be used, and where the supply of water is limited. This device consists of a radiator through which the cooling water is forced as it issues from the engine. It is made up of a large number of small tubes having radiating flanges around them or of other suitable design, affording a large cooling surface. A fan operated by electric motor is placed in front of the radiator, as shown in the illustration, and is arranged to furnish a strong current of air passing through the various coils of the radiator, taking up the heat of the water in the tubes and quickly cooling same. The power required by the motor is approximately 10% of the power developed by the engine. A difference in temperature can be obtained between the inlet and outlet water when using this device of from 25° to 30° Fahr.

About 40 gallons of water should be circulated through the coils per actual horse-power per hour. These figures, however, depend upon the design of the radiator and the conditions of temperature under which it is to operate.

On account of the large amount of power absorbed by the motor, this outfit is only suitable for special installations where other cooling methods cannot be used.

COOLING TOWERS

Where cooling tanks cannot be installed, for instance in large installations where enormous capacity of tanks would be required, a cooling tower as shown at Fig. 48 and Fig. 48*a* can be advantageously used. In this case, the heated water as it issues from the engine cylinder water-jacket is pumped to the top of the cooling tower, which is placed in a position to allow of the best cooling effect, the water simply flowing down the surfaces of the cooling tower, and its temperature being reduced by coming in contact with the air. Where large amounts of water have to be cooled, a fan is added to increase the draught of air coming in contact with the water to be cooled.

Exhaust Silencers.—The noise from the exhaust gases is sometimes considered to be a great objection to the use of explosive engines, but this is chiefly due to the fact that the ordinary cast-iron exhaust silencing chamber supplied with engine is not designed to entirely silence the exhaust, but is only regarded as sufficient to partly reduce this noise.

Where it is essential that the exhaust be entirely silenced, this can be easily accomplished in the following way: A brick pit should be built as shown in Fig. 49. The exhaust-pipe from the engine is then connected to the bottom of this pit. The outlet-pipe to the atmosphere is connected to the top of the pit. The space inside the pit should be filled with large stones, as shown in illustration. These stones should be about six inches in size, so that crevices are left

between them through which the gases can penetrate. A drain-pipe should be arranged to allow the water to flow out of the pit. The stone or cast-iron plate covering the pit is securely fastened down to the masonry.*

With oil-engine exhaust gases there may be some odor. When it is necessary that both the noise and the

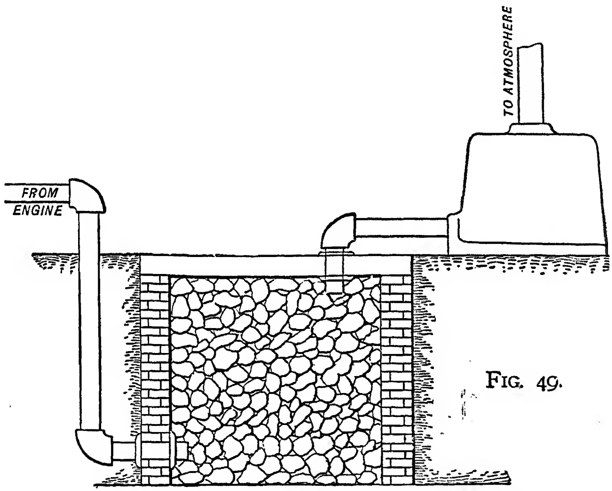


FIG. 49.

odor should be done away with, an exhaust washer should be installed instead of the silencing pit, as already described. This apparatus consists of a tank, to which the water is connected as it issues from the water-jacket of the engine-cylinder, or where cooling

*In some cases the connection is made direct from the engine to the silencer, and thence to the pit, the exhaust pipe leading to the atmosphere being supported from the covering over the pit.

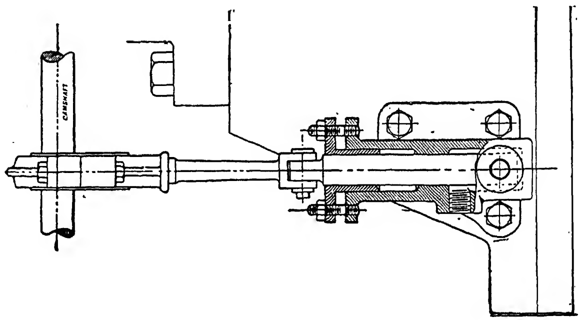

CAMSHAFT

FIG. 50.

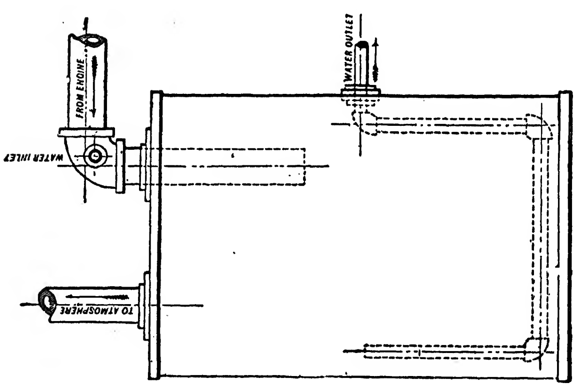

FROM ENGINE
WATER INLET
WATER OUTLET
TO ATMOSPHERE

FIG. 51.

tanks are used the water should be taken from the main. About 100 gallons of water are required per hour. The exhaust-pipe from the engine valve-box is also connected directly to this tank. The outlet of the water is connected from the tank to sewer and the outlet exhaust-pipe is also connected in the usual way to the top of the building.

The exhaust gases by this arrangement come in contact with the water and are partly condensed and quite purified. The pressure and noise are eliminated entirely, any deposit of carbon left in the gases after combustion is carried off by the water to the sewer, and there is practically no odor when the gases escape from the exhaust-pipe to the atmosphere at the roof. This device is shown in Fig. 51. The sizes given for piping and tank are those suitable for a 10 to 20 H. P. oil engine. The internal piping in the tank is so placed to avoid any pressure which is created inside the tank due to the exhaust gases of the engine from entering the sewer. If any water is blown out at the top of the exhaust-pipe, a steam exhaust-head is used for obviating this. This apparatus is the same as used on steam exhaust-pipes.

Sizes for piping and tank for a 10 to 20 H. P. oil engine:

Pipe from engine, 3″ diameter.
Pipe of water inlet, $\frac{3}{4}$″ diameter.
Pipe to atmosphere, 3″ diameter.
Pipe to water outlet, 2″ diameter.
Size of tank, 2′ in diameter by 4′ high.

When it is required to partly silence the noise of exhaust only part or all of the water from the cooling jacket can be turned into the exhaust-pipe directly from the water-jacket. The water is allowed to run to waste again at the silencer. (See Fig. 52.) Wherever water is connected to the exhaust-pipe, care must be taken that none can under any condition enter through

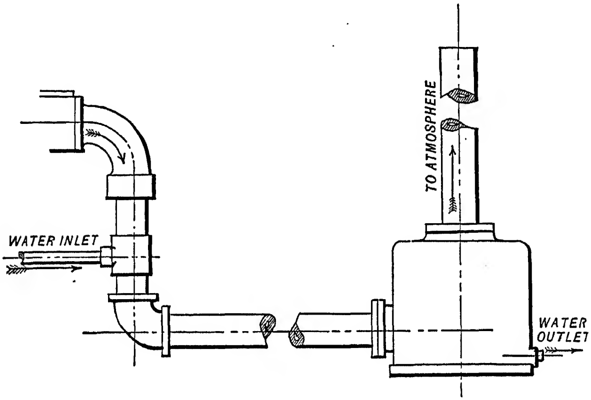


FIG. 52.

the exhaust valve-box into the cylinder or vaporizer of the engine. Where water enters the silencer or the piping under pressure from the city main or otherwise, it is necessary that the area of the outlet-pipe be large enough to allow the water to drain freely at atmospheric pressure. If the water is not allowed free drainage, it may quickly fill up the silencer, and perhaps enter the valve-box of the engine, causing the engine to stop working.

SELF-STARTERS.—Engines of 25 H. P. and over should be provided with separate means of starting besides the relief-cam for reducing the pressure of compression as usually provided with the smaller sizes of engines. The weight of the fly-wheels and reciprocating parts on the larger engines which are to be put in motion when being started necessarily entails considerable exertion, and the strength of two men is required to do this work where no other means is provided for this purpose.

There are several different self-starting devices made for gas engines, and it is much easier to accomplish this work with a gas than with an oil engine, since with the former gas only has to be dealt with and can be readily diluted with air and an explosive mixture formed, whereas with the oil engine the fuel must be vaporized first and then mixed with the air before an explosive mixture is available to be ignited and the impulse on the piston obtained. In order, therefore, to accomplish these various operations necessary in the oil engine, sufficient power must be independently provided to turn the engine crank-shaft over two or three revolutions so that the different mechanisms can work, the fuel be injected or inducted into the cylinder or vaporizer, become mixed with the incoming air and an explosion obtained, thus giving the required impulse. This power is usually derived from a separate air reservoir charged during the previous running of the engine or from a small air-compressor operated by hand.

The self-starter used with the Hornsby-Akroyd type

of oil engine is shown in Fig. 53. The reservoir is connected to air and exhaust valve-box of engine through a supplementary valve-box containing two check-valves. These check-valves are arranged to be lifted from their seats by means of the hand-lever as shown.

The following are the instructions in detail for starting these engines by means of this device. (These re-

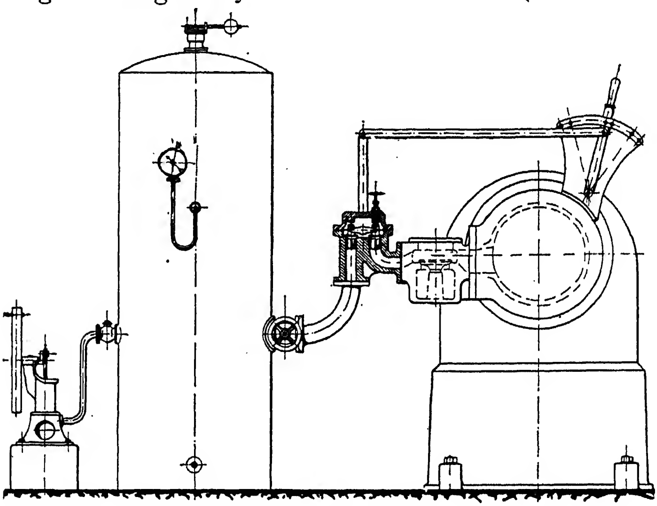


FIG. 53.

marks are generally applicable to all types of engines provided with starting devices of this principle.)

See that the valve *A* on the steel receiver is open, and also the cock *B* on the pipe leading from the hand air-pump. Put the starting lever in the quadrant at the position marked "Running and when charged," and pin it there. Then screw down the valve *C* on the double valve-box, and pump air into the receiver by the

air-pump up to a pressure of say 60 or 70 lbs. to the square inch as shown on the gauge. Then close the cock *B* on the air-pump pipe, withdraw the pin in the starting lever, and put it in the hole by the side of the lever to act as a stop; then place the engine ready for starting as elsewhere described. Place the crank a little over the dead centre in whichever direction the engine is intended to run, unscrew the valve *C* in double valve-box, and then suddenly push the starting lever forward to the end of the quadrant, and the engine will start. Pull the lever back immediately against the pin, and screw down the valves on the double valve-box and on the receiver. Before stopping the engine at any time, pull the lever back and pin it in hole marked "To charge;" unscrew the valves on the double valve-box and receiver, and allow the engine to pump air into the receiver again to 80 or 100 lbs. pressure; put the lever to the centre hole marked "When running, and when charged," and pin it there; screw down the valves on the receiver and valve-box, and the air pressure in the receiver will be retained in readiness to start the engine the next time it is required. If an air-pump is not provided, the engine must be started in the usual way the first time, by pulling round the fly-wheel, and the receiver afterward filled each time before stopping.

THE UTILIZATION OF WASTE HEAT FROM OIL ENGINES.—With many installations of oil engines, the question of utilizing the waste heat from the water-jacket and exhaust gases is considered. The amount of heat lost in this way of course varies with different

types of engines according to their thermal efficiency. Reference to the following table shows the amount of heat rejected in the cooling water and exhaust.

The two greatest disadvantages to the utilization of waste heat are: *First,* the oil engine furnishes heat only when in operation, and therefore a separate heater is required to furnish the necessary heat when the engine is stopped; and *secondly,* as the exhaust gases from most oil engines are not clean, accumulation of carbon results in the passages through which the heated gases pass and necessitates frequent cleaning.

HEAT BALANCE PER ACTUAL OR B. H. P. PER HOUR.

	B. T. U.		B. T. U.
Received by engine 0.8 lb. of fuel at 19,000 B. T. U. per lb. 19,000 × 0.8 lb. =	15,200	Heat equivalent shown on brake (82% mech. ef.)	3,104
		Heat lost to jacket water 47.4%	7,200
		Heat lost to exhaust 25%.	3,800
		Lost in radiation and unaccounted for.	1,096
	15,200		15,200

The above table is based on 0.8 lb. fuel consumption per actual H. P. hour. With engines having a higher economy, the amount of heat rejected would be reduced. Assume the efficiency of the heating appa-

ratus to be 68%, then with the heat rejected by the water jacket, viz., 11,000 B. T. U., 7,480 B. T. U. should be available for heating purposes per actual H. P. per hour.

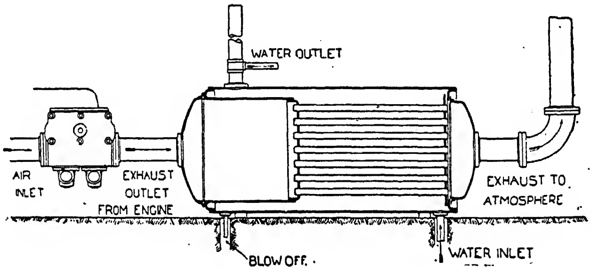


FIG. 54.

An apparatus designed to utilize the waste heat from the exhaust is shown at Fig. 54. The heat could be utilized either by water circulation or by means of heated air, a blower being used to pass the cold air over the heated water pipes or by steam heat direct. With the first arrangement piping in which the water is circulated would have to be of sufficient length to allow the water to give out its heat. With the second arrangement (that of heated air) sufficient quantity of air should be passed over or through the piping in which the heated water flows. This heated air is then passed through ducts to the spaces to be heated in the ordinary way. The third system, namely, steam heat, would require the exhaust gases to raise the temperature of the water above the boiling point, 212°. Each pound of steam at 212° evaporated from water at 140° requires 1038 B.T.U. As previously stated, if the

efficiency of the heating apparatus is as high as 68%, then there is available from the exhaust gases.

$3800 \times 0.68 = 2584$ B.T.U. per B.H.P. per hour.

This heat will be sufficient to raise about 2½ lbs. of water to 212° steam or somewhat less than this amount to steam at 15 lbs. gauge pressure. It is estimated that 3.6 B.T.U. are required to maintain a cubic foot of space at 70° F. when the weather is at zero outside, and 2.6 B.T.U.'s are required to maintain the same temperature inside when the outside temperature is 20° F. These figures, of course, have to be varied with different buildings. The above figures are also estimated with the engine running at full load. At half load only about 60% of the heat above referred to would be available.

EXHAUST TEMPERATURE.—The temperature of the exhaust gases is difficult to ascertain correctly. The temperature of the exhaust from the Diesel engine is recorded by Professor Denton as being approximately 740° Fahr. The temperature of different oil-engine exhaust gases varies, and it is probably considerably above that figure. This temperature varies also, of course, according to the size of the engine, and also according to the power that the engine is developing. The heat is greatest at full load and on the largest engines.

CHAPTER V.

OIL ENGINES DRIVING DYNAMOS.

OIL ENGINES for many reasons are well adapted for driving dynamos generating electric current in isolated lighting plants. A large number of such installations have been made in recent years. The oil engine is self-contained, and, unlike a gas engine, is independent of gas works or gas-producer plant for its supply of fuel. Small power installations with oil engines as prime movers should require also less attention than a plant equipped with steam engine and boilers. There is probably not the danger there is with a steam engine of explosion, and as the fuel used is ordinary kerosene of a safe flashing point, there can be little or no fear of destruction by fire. Practically, no hauling of fuel is required, nor is there, with an oil engine, any consumption of water if storage tanks are installed. Further, an oil engine does not deteriorate if only required for part of the year and left standing idle for the remainder of the time. With these and, perhaps, other advantages possessed by oil engines, their adaptability for driving dynamos in isolated electric-lighting and power plants may be understood. Fig. 55 illustrates an oil

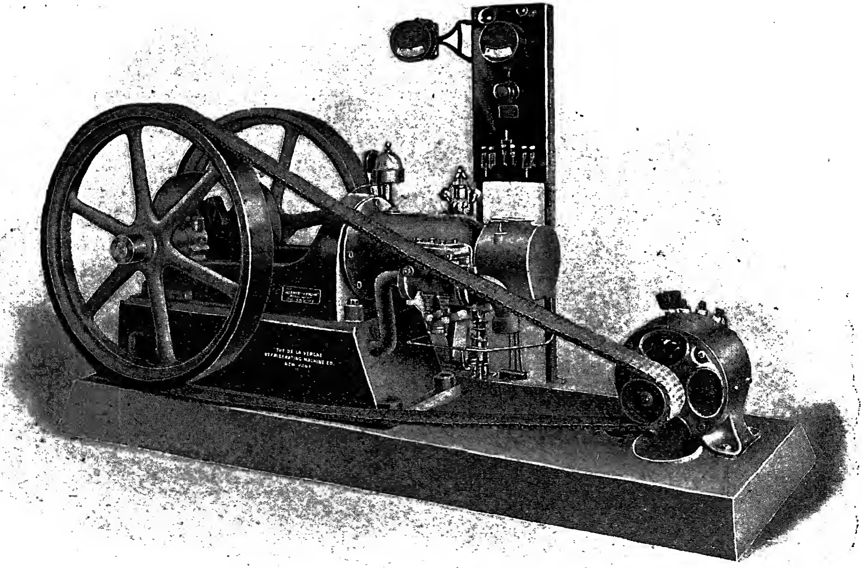


Fig. 55.

engine driving dynamo with link belt. The dynamo is placed close to the engine to economize floor space.

This plant is arranged with the cams having been set for the engine to run backwards.

INSTALLATION.—In order that the plant may be entirely satisfactory and give the best results, it is very essential that the engine and dynamo be correctly located with regard to each other and properly installed at the outset.

THE FOUNDATIONS both for the engine and for the dynamo should be built of good cement concrete, and should be placed on solid ground, so that they are steady and without vibration. The engine foundation can be made as shown at Fig. 56. When, however, the ground that the foundation is built upon is not solid, it is preferred to build the foundation more tapered than shown toward the bottom, thus increasing the surface that the concrete rests on. The weight of the foundation is considered sufficient allowing about 5 cubic feet per I. H. P. for engines under 50 H. P. for concrete. For engines over 50 I. H. P. the foundation can be reduced per I. H. P. If the foundation is built of brickwork, its dimensions should be somewhat greater than those given for concrete. The ingredients of the best concrete are broken stone, Portland cement and sharp sand. The fuel tank placed underground surrounded with concrete and installed in accordance with the requirements of the fire underwriters is shown at Fig. 56*a*. The fuel supply pipe connections and fuel supply pump are also shown as required by their regulations.

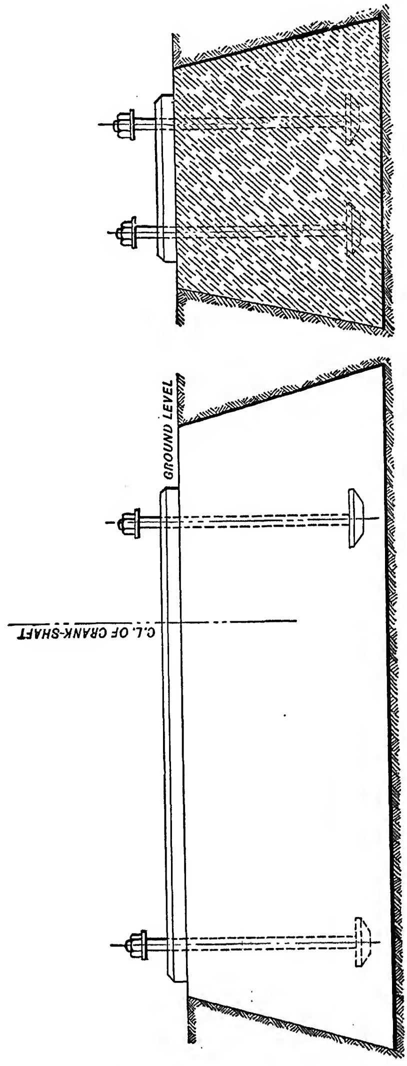


Fig. 56.

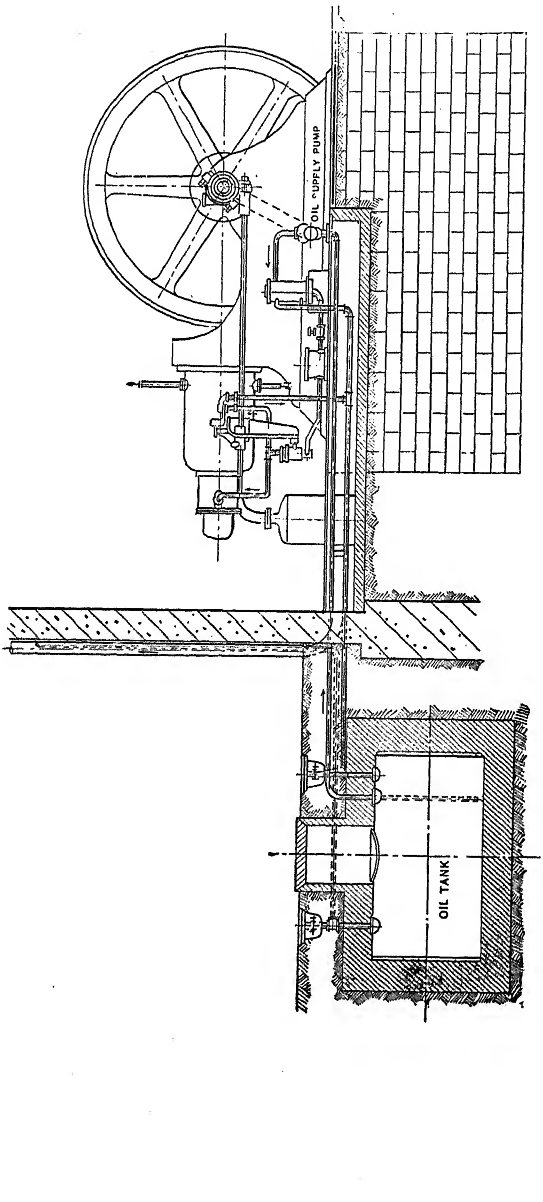


FIG. 56a.

When driving by belt the distance between the centres of the dynamo and the engine-shafts is an important feature. Where space is restricted and it becomes essential that the dynamo be placed as close as possible to the engine, it is advantageous to use a link leather belt, allowed to run quite loose, the part of the belt in tension being underneath, the loose part being on top, so that the arc of contact made on the smaller pulley of the dynamo is as great as possible. This arrangement with loose belt lessens the friction on the bearings, which would be occasioned if the belt were made tight, as required at short centres with ordinary leather belt. When using link leather belt, the distance between the centres should be with the usual standard size of fly-wheels 2 to 2.5 diameters of the engine fly-wheels—that is, the distance should not be less than 7 ft. for wheels of 3′ 6″ diameter and not greater than 15 ft. for wheels of 6 ft. diameter. Where ordinary leather belt is used instead of link belt, this distance should be increased to 3 diameters of fly-wheel, but in any case this dimension should not exceed 18 ft. for driving wheels 6 ft. in diameter. To obtain absolutely steady light, it is sometimes advantageous to place a balance-wheel on the armature shaft of the dynamo. This wheel if used should weigh about 15 lbs. per K. W. of dynamo, and be of such diameter that at the maximum speed of dynamo its peripheral speed will not exceed 6000 ft. per minute. This wheel must be accurately balanced, and is usually cast in one piece with pulley, as shown in Fig. 57. The

necessary width of belt to transmit the H. P. may be calculated as follows:

$$\text{H. P.} = \frac{V \times w}{800}.$$

H. P. = the actual horse-power.
V = velocity of belt in feet per minute.
w = width of belt in inches.

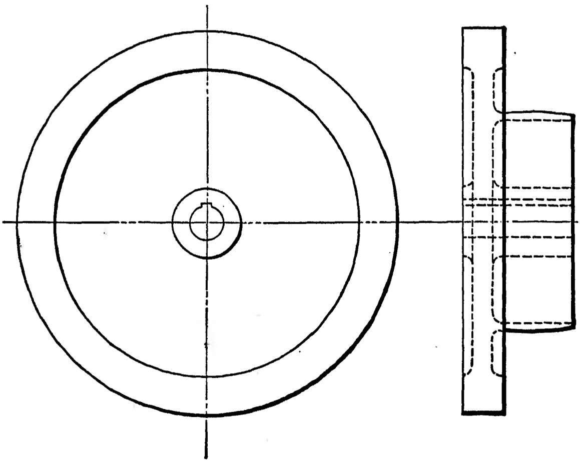


FIG. 57.

The maximum number of incandescent lights available from the dynamo per brake or actual H. P. of engine varies according to the efficiency of the dynamo, and the efficiency of the means of transmission as well as to the efficiency of the electrical installation. Lack of

power as recorded by the electrical instruments is not necessarily due only to defects of the engine, as leakage of power may occur in various ways, as above stated. Usually ten 16 candle-power lights per Brake H. P. are calculated as being a fair load for the engine. With arc lamps of 2000 candle-power, the B. H. P. of engine for each lamp required is approximately .75. It is advisable to have spare power with an explosive engine above that required to run all the lights. Losses of power should be allowed for in the belt, which vary from 10 to 15 per cent.

The regulation of explosive engines for electric lighting must necessarily be such that there is no flicker in the incandescent lights. A speed variation of 2 per cent. is now guaranteed with several oil engines. This regulation gives a very good light and equals that developed with many steam engines.

When space is not available to permit the use of belt transmission, the dynamo is connected directly on to the shaft of the engine, as in Figs. 58 and 58*a*. The coupling between engine-shaft and dynamo is usually flexible to allow of dynamo bearings and the engine-shaft bearings remaining in alignment when they become worn. In direct-connected plants the loss due to the belt transmission is avoided, and a saving is thus effected; but, on the other hand, the first cost of the dynamo is very much greater, running, as it does, at a slower speed than the belt-driven machine, and therefore is of larger dimensions, and consequently more costly.

Fig. 58 illustrates a Hornsby-Akroyd engine of the

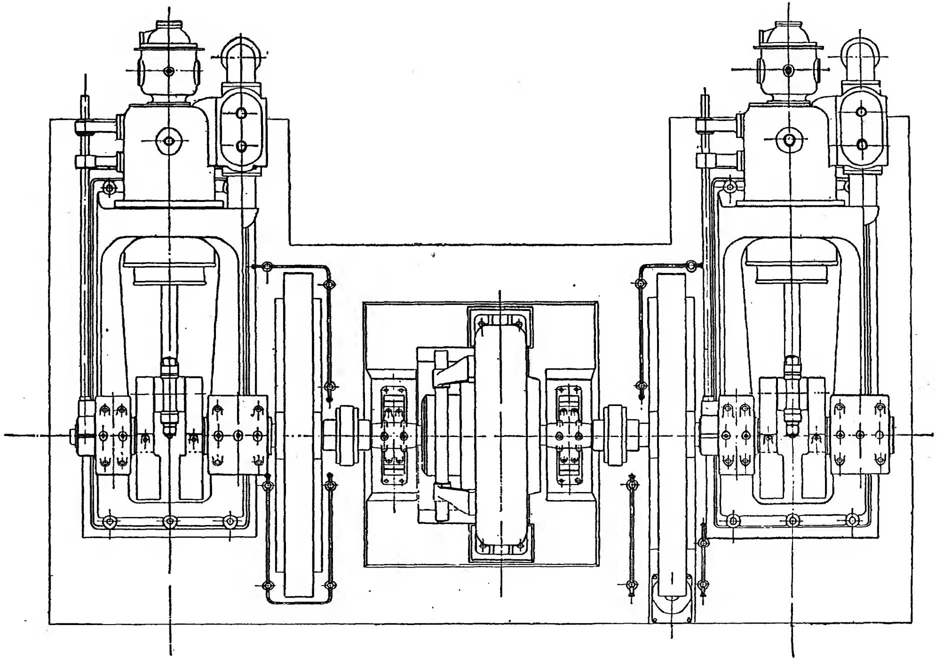


FIG. 58.

twin cylinder horizontal type coupled direct to the generator. The illustration shows the engines placed each side of the generator with two flywheels and connected by coupling forged on the shaft. An arrangement preferred is the two engines placed side by side with one heavy flywheel, the generator is coupled to the engine shaft and placed on one side. Where this outfit has been used for power purposes the timing of the air inlet and exhaust cams has been such that the explosions have been simultaneous in each cylinder. In this way the strain on the generator shaft has been reduced.

Fig. 58*a* illustrates the Mietz & Weiss horizontal type of engine directly connected to dynamo through flexible coupling. This engine, being of the two-cycle type, receives an impulse at each revolution of the crank-shaft, and it runs very regularly and at a high rotative speed.

The method of working of the Mietz & Weiss engine is fully described in Chapter IX.

The fly-wheels of explosive engines intended for driving dynamos are usually made heavier than when the engines are required for other purposes. (See Chapter II.)

Notwithstanding the special design of engines for electric-lighting purposes and apparent correct adjustment of the governing mechanism, the lights may sometimes be seen to flicker. Flickering in the incandescent lights can be easily located by close inspection of the engine and dynamo, and may be due either to the fly-wheels, the governor, the belt, or the dynamo itself. To precisely locate this defect and remedy it,

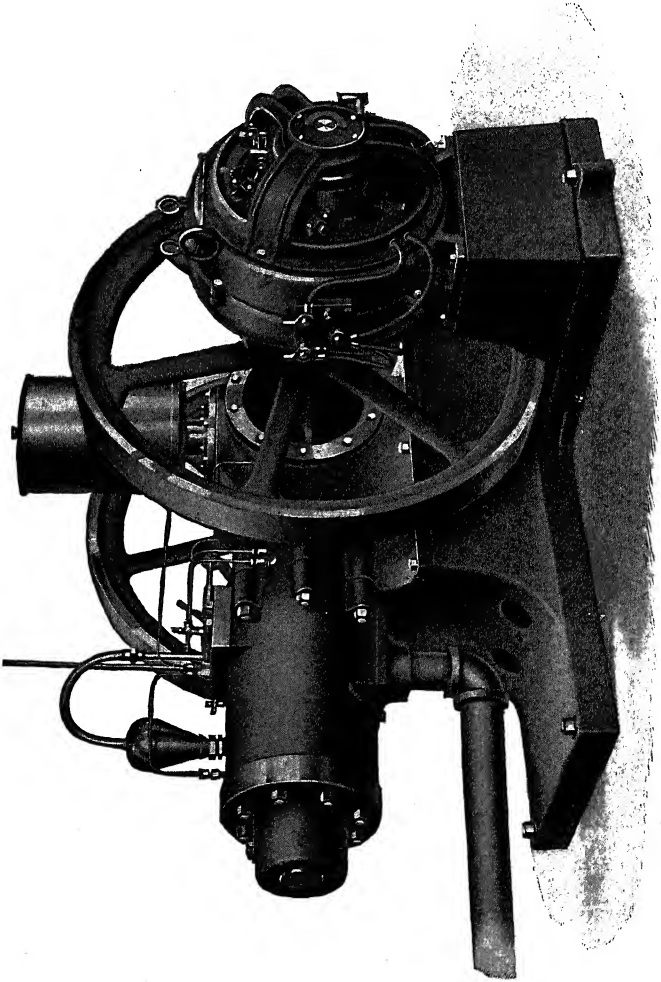


Fig. 58a

notice the lamps carefully. If the variations in the light are due to want of fly-wheel momentum, such variations will be seen to coincide with the number of revolutions of the engine. Again, if the variation in the lights is only periodical, then this defect should be remedied by adjustment of the governor. Examine carefully the governing mechanism of the engine. If the variation is caused by the governor acting too slowly, then adjust so as to cause more rapid contact with the valve or other controlling mechanism.

The cause of the trouble may not be, as already suggested, in the fly-wheel momentum or in the adjustment of the governor, but in the belt, which is frequently the sole cause of unsatisfactory lighting. The engine and dynamo pulleys over which the belt runs must be exactly in line with each other. The belt should be endless, or if jointed such joints should be very carefully made. A thick, uneven joint in the belt will cause a flicker in the lights each time it passes over the dynamo pulley. The belt should be allowed to run as loose as possible. The writer has seen belts running quite slack and most satisfactorily when the pulleys have been covered with specially prepared pulley-covering material. In some instances in the dynamo itself may be found the cause of the variation in the voltage. If the commutator becomes unevenly worn, or if the brushes are not properly adjusted, unsteady lights will result, and then the commutator should be made of even surface and the brushes correctly adjusted.

Oil engines can be stopped if desired by pressing button in the dwelling-house, an attachment being

added to some engines which automatically turns the stopping handle. This is an advantage where the light is required late at night, and allows the attendant to leave the engine early, at the same time providing requisite illumination as long as required.

Air Suction.—The noise created by the air being drawn into the cylinder has, in some cases, to be silenced. This can be accomplished by connecting the air-inlet pipe to wooden box containing space at least five times as great as the volume of the cylinder—the sides of the box having holes which are lined with rubber. The total area of all these small inlet air holes should be at least three times the area of the air-inlet pipe to the engine.

CHAPTER VI.

OIL ENGINES CONNECTED TO AIR-COMPRESSORS, PUMPS, ETC.

THE use of compressed air is now being extensively applied as a means of power transmission, and it is coming more and more into favor in this connection also for actuating pneumatic tools, and for other purposes too numerous to mention. Many advantages are claimed for the combination of explosive engines connected to air-compressors as a motive power.

Skilled attention is not necessary at all times. There are practically no standby losses, and the outfit can be easily transported. A small size compressor is shown in section at Fig. 59*a* made by the Bury Mfg. Co., Erie, Pa. The normal speed of these compressors being considerably less than the normal speed of oil engines, they are operated by gearing or by belt from the engine.

Fig. 60 shows an oil engine geared to air-compressor of the ordinary double-acting type. In this outfit the power necessary to actuate the compressor is transmitted by gearing from the engine crank-shaft to the compressor-shaft, which then revolves at a slower speed than the engine-shaft. This arrangement is con-

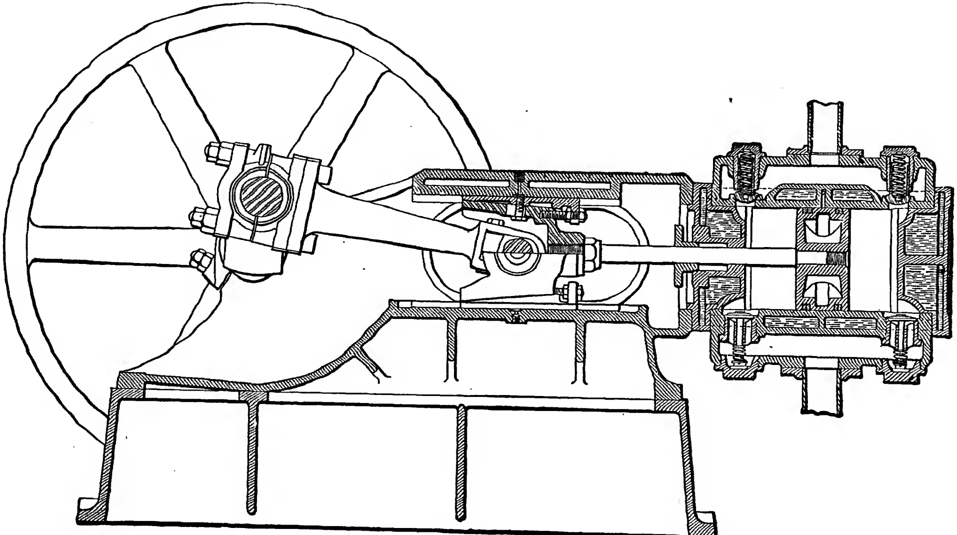


FIG. 59a.

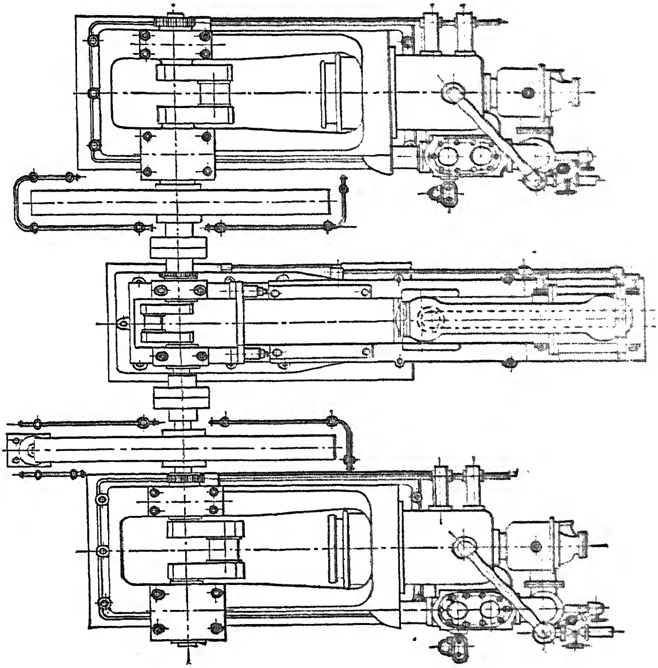


FIG. 59.

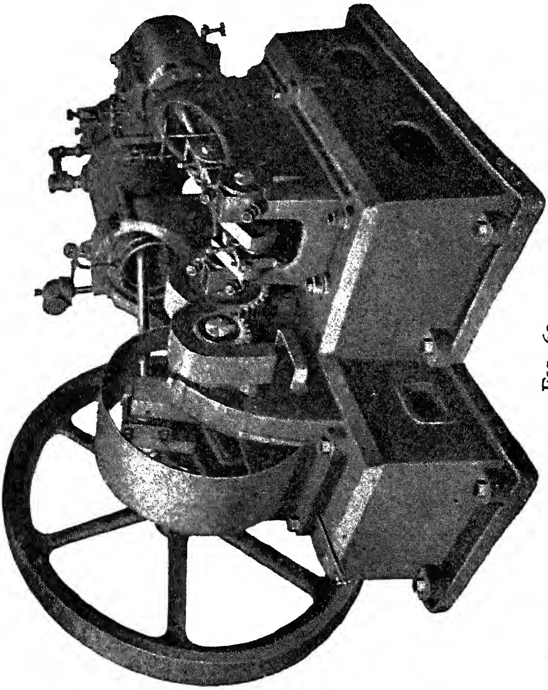


Fig. 60.

sidered advantageous, because of the slower motion of the air-compressor valves as compared with the direct-connected outfit. In each of the illustrations the air-compressor cylinder is water-jacketed, the circulating water being supplied by the small pump actuated from the engine cam-shaft, the water being first delivered to the compressor cylinder, and thence to the oil engine cylinder. This outfit consists of 13 B. H. P. oil engine and "Ingersoll-Sergeant" double acting air-compressor having cylinder 8″ diameter and 8″ stroke, and running at 150 revolutions per minute, delivering 70 cubic ft. of free air per minute at 70 to 80 lbs. pressure.

The horse-power required to operate a compressor delivering an actual amount of air at a given pressure can be found from the diagram at Fig. 60*c*. The theoretical horse-power required to compress 100 cubic feet, delivered at various pressures up to 125 lbs. can be taken directly from the curves on this diagram.

In order to find the actual horse-power, the indicated efficiency and the mechanical efficiency of the compressor should be known. The indicated efficiency is the relation of the theoretical working diagram to the real indicated power. In the curve (Fig. 61*a*), the actual air delivered is given. Approximately 10% should be added to allow for losses due to heating of the air, valve resistance and friction.

Fig. 59 shows a 250 H. P. oil engine of the horizontal type direct connected to a two-stage air compressor in which the low pressure cylinder is 20½ inches diameter, and the high pressure cylinder 13½ inches, and is designed to furnish 1,275 cubic feet at 90 lbs. pressure per minute.

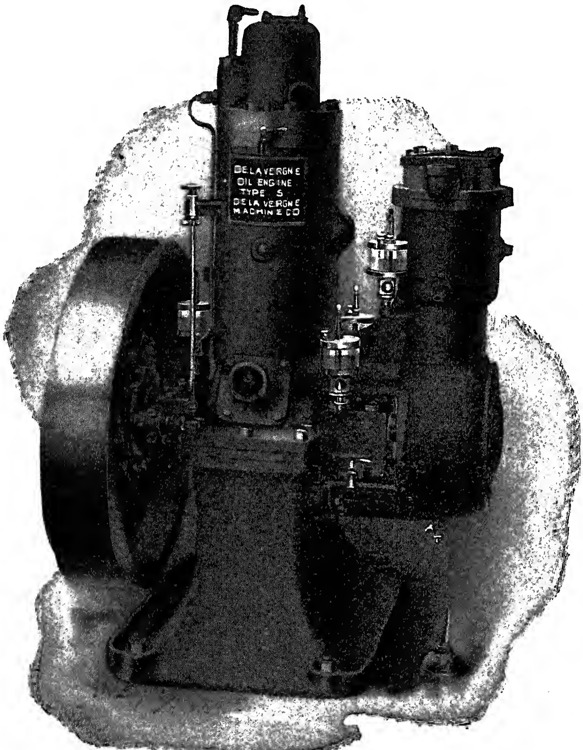
TABLE II.—VARIOUS AIR PRESSURES.—RICHARDS'.

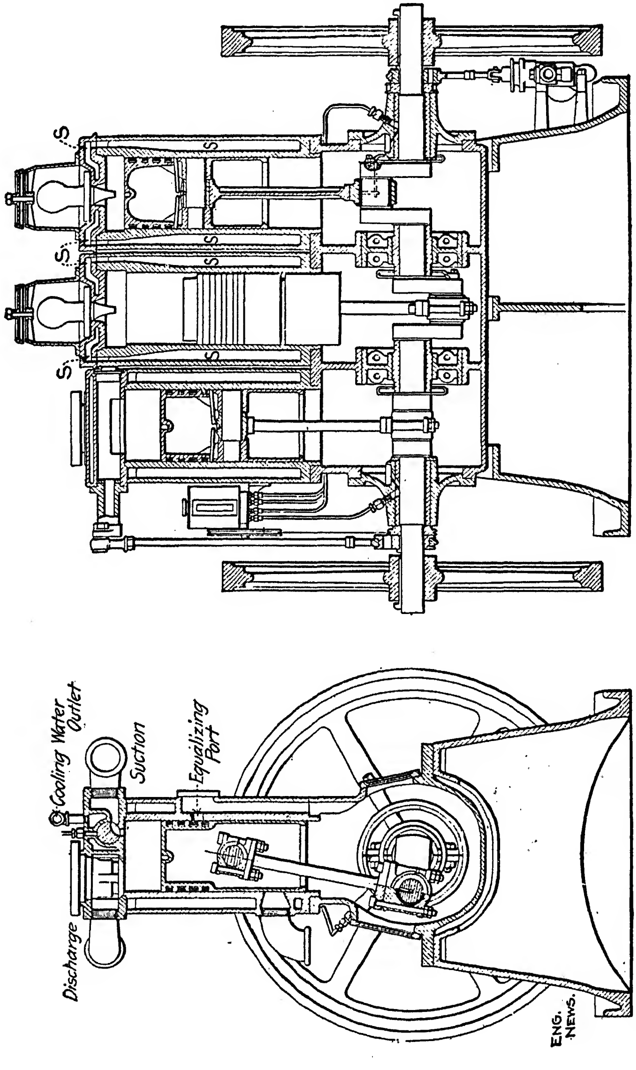
Gauge Pressure.	Absolute Pressure.	Pressure in Atmospheres.	Volume with Air at Constant Temperature.	Volume with Air Not Cooled.	Mean Pressure per Stroke. Air Constant Temperature.	Mean Pressure per Stroke. Air Not Cooled.	Mean Pressure during Compression only. Air Constant Temperature.	Mean Pressure during Compression only. Air Not Cooled.	Final Temperatures. Air Not Cooled.	Gauge Pressure.
0	14.7	1	1	1	0	0	0	0	60	0
1	15.7	1.068	.9363	.95	.96	.975	.43	.44	71	1
2	16.7	1.136	.8803	.91	1.87	1.91	.95	.96	80.4	2
3	17.7	1.204	.8305	.876	2.72	2.8	1.4	1.41	88.9	3
4	18.7	1.272	.7861	.84	3.53	3.67	1.84	1.86	98	4
5	19.7	1.34	.7462	.81	4.3	4.5	2.22	2.26	106	5
10	24.7	1.68	.5952	.69	7.62	8.27	4.14	4.26	145	10
15	29.7	2.02	.495	.606	10.33	11.51	5.77	5.99	178	15
20	34.7	2.36	.4237	.543	12.62	14.4	7.2	7.58	207	20
25	39.7	2.7	.3703	.494	14.59	17.01	8.49	9.05	234	25
30	44.7	3.04	.3289	.4638	16.34	19.4	9.66	10.39	255	30

35	49.7	3.381	.2957	.42	17.92	21.6	10.72	11.59	281	35
40	54.7	3.721	.2687	.393	19.32	23.66	11.7	12.8	302	40
45	59.7	4.061	.2462	.37	20.52	25.59	12.62	13.95	321	45
50	64.7	4.401	.2272	.35	21.79	27.39	13.48	15.05	339	50
55	69.7	4.741	.2109	.331	22.77	29.11	14.3	15.98	357	55
60	74.7	5.081	.1968	.3144	23.84	30.75	15.05	16.89	375	60
65	79.7	5.423	.1844	.301	24.77	31.69	15.76	17.88	389	65
70	84.7	5.762	.1735	.288	26.	33.73	16.43	18.74	405	70
75	89.7	6.102	.1639	.276	26.65	35.23	17.09	19.54	420	75
80	94.7	6.442	.1552	.267	27.33	36.6	17.7	20.5	432	80
85	99.7	6.782	.1474	.2566	28.05	37.94	18.3	21.22	447	85
90	104.7	7.122	.1404	.248	28.78	39.18	18.87	22.	459	90
95	109.7	7.462	.134	.24	29.53	40.4	19.4	22.77	472	95
100	114.7	7.802	.1281	.232	30.07	41.6	19.92	23.43	485	100
105	119.7	8.142	.1228	.2254	30.81	42.78	20.43	24.17	496	105
110	124.7	8.483	.1178	.2189	31.39	43.91	20.9	24.85	507	110
115	129.7	8.823	.1133	.2129	31.98	44.98	21.39	25.54	518	115
120	134.7	9.163	.1091	.2073	32.54	46.04	21.84	26.2	529	120
125	139.7	9.503	.1052	.202	33.07	47.06	22.26	26.81	540	125
130	144.7	9.843	.1015	.1969	33.57	48.1	22.69	27.42	550	130
135	149.7	10.183	.0981	.1922	34.05	49.1	23.08	28.05	560	135
140	154.7	10.523	.095	.1878	34.57	50.02	23.41	28.66	570	140
145	159.7	10.864	.0921	.1837	35.09	51.	23.97	29.26	580	145
150	164.7	11.204	.0892	.1796	35.48	51.89	24.28	29.82	589	150
160	174.7	11.88	.0841	.1722	36.29	53.65	24.97	30.91	607	160
170	184.7	12.56	.0796	.1657	37.2	55.39	25.71	32.03	624	170
180	194.7	13.24	.0755	.1595	37.96	57.01	26.36	33.04	640	180
190	204.7	13.92	.0718	.154	38.68	58.57	27.02	34.06	657	190
200	214.7	14.6	.0685	.149	39.42	60.14	27.71	35.02	672	200

The outfit runs at 150 R. P. M. The crank-shafts of the engine are coupled to the crank-shaft of the air compressor by means of couplings forged on the end of the shafts. In this case the explosions in the engine are timed to take place simultaneously.

Fig. 60*b* shows a vertical Mietz & Weiss oil engine direct connected to a single acting high speed air compressor. The engine operates on the two-cycle plan,





Longitudinal Section of Unit.

Cross Section of Air Compressor.

FIG. 60b.

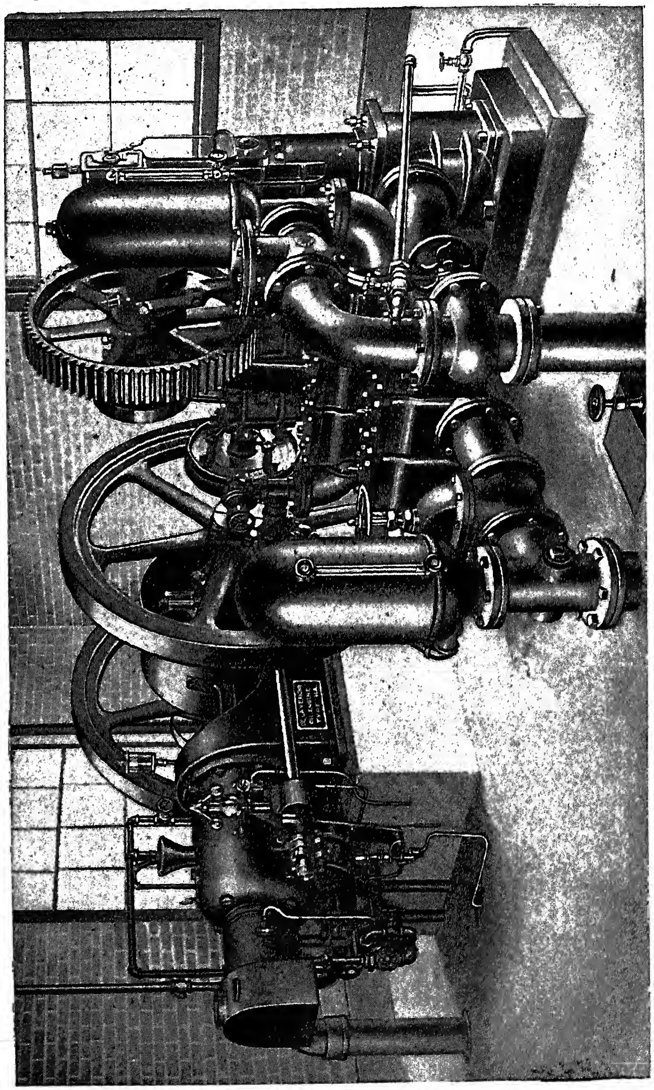


Fig. 61

similar to that explained on page 178. It runs at 420 R. P. M. Diameter of the air-compressor cylinder is 8″ and the stroke 8″. The piston displacement being approximately 97 cubic feet of free air per minute.

Another direct connected high speed type of air compressor is that shown at Fig. 60*a*, consisting of a De La Vergne Type S oil engine of the two-cycle, vertical type direct connected to a single acting compressor actuated directly from the crank-shaft of the engine and running at the same speed, namely, 450 to 500 R. P. M. The valves of this compressor are of special design, being simply a sheet-steel plate specially adapted for running at this high rate of speed. These outfits are made up to 25 H. P.

TABLE III.—EFFICIENCIES OF AIR-COMPRESSORS AT DIFFERENT ALTITUDES.

Altitude, feet.	Barometric, Pressure.		Volumetric Efficiency of Compressor, Per Cent.	Loss of Capacity, Per Cent.	Decreased Power Required, Per Cent.
	Inches, Mercury.	Pounds Per Square Inch.			
0	30.00	14.75	100.	0.	0.
1000	28.88	14.20	97.	3.	1.8
2000	27.80	13.67	93.	7.	3.5
3000	26.76	13.16	90.	10.	5.2
4000	25.76	12.67	87.	13.	6.9
5000	24.79	12.20	84.	16.	8.5
6000	23.86	11.73	81.	19.	10.1
7000	22.97	11.30	78.	22.	11.6
8000	22.11	10.87	76.	24.	13.1
9000	21.29	10.46	73.	27.	14.6
10000	20.49	10.07	70.	30.	16.1
11000	19.72	9.70	68.	32.	17.6
12000	18.98	9.34	65.	35.	19.1
13000	18.27	8.98	63.	37.	20.6
14000	17.59	8.65	60.	40.	22.1
15000	16.93	8.32	58.	42.	23.5

The efficiency of an air compressor is reduced when working at high altitudes. Table III. gives such depreciation in efficiency at the different altitudes.

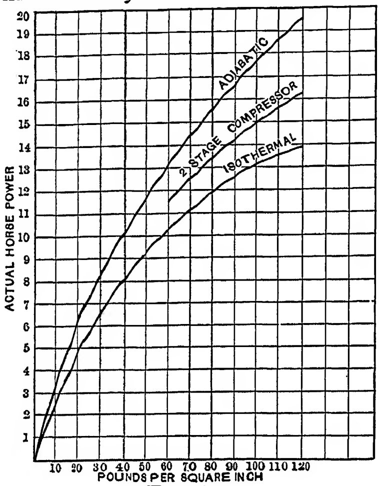


FIG. 61*a*.

OIL PUMPING STATIONS

Fig. 61*b* shows the oil engine connected by friction coupling directly with a Goulds triplex power pump. The illustration shows a complete pumping station used in the oil fields for transporting crude oil from the oil fields to the oil refinery. Pressures as high as 900 to 1,000 lbs. are frequently used in this work and it is customary for the engines to operate 24 hours per day continuously. The illustration shows several outfits, one of which is at all times held in reserve. This illustration is given to show one of the many applications of the oil engine used in connection with a pump. In these

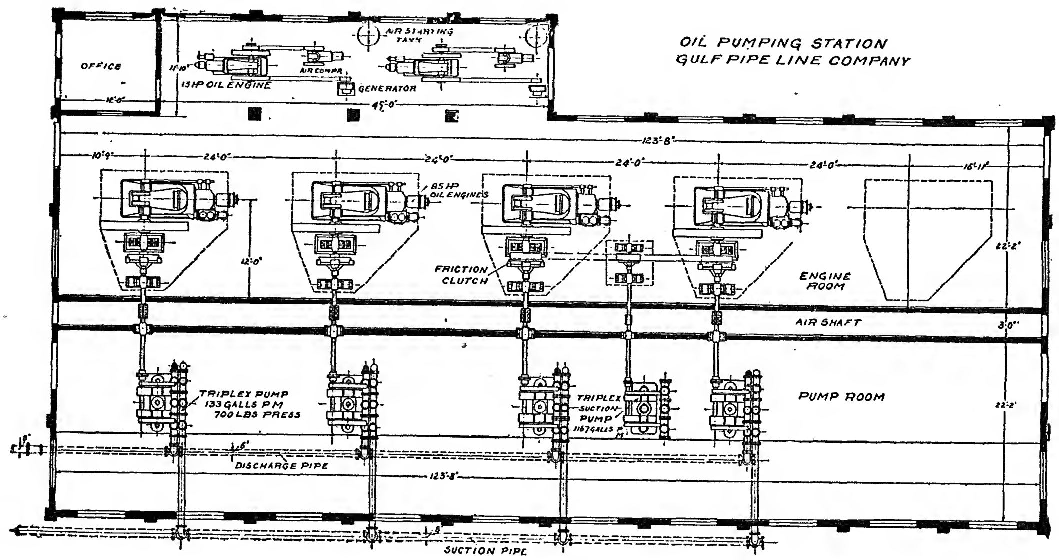

OIL PUMPING STATION
GULF PIPE LINE COMPANY
OFFICE
AIR STARTING TANK
AIR COMPR.
13 HP OIL ENGINE
GENERATOR
85 HP OIL ENGINES
FRICTION CLUTCH
ENGINE ROOM
AIR SHAFT
PUMP ROOM
TRIPLEX PUMP 133 GALLS P.M. 700 LBS PRESS
TRIPLEX SUCTION PUMP 1167 GALLS P.M.
DISCHARGE PIPE
SUCTION PIPE
123'-8"
24'-0"
10'-9"
16'-11"
22'-2"
3'-0"
12'-0"
49'-0"

FIG. 61*b*.

cases, the engine operates on crude oil, which is passed through the pipe line and effects great economy as compared with the steam plant. The oil engine is now very largely used for this purpose.

OIL-ENGINE PUMPING PLANTS.—Fig. 61 represents an oil-engine pumping plant as installed for supplying

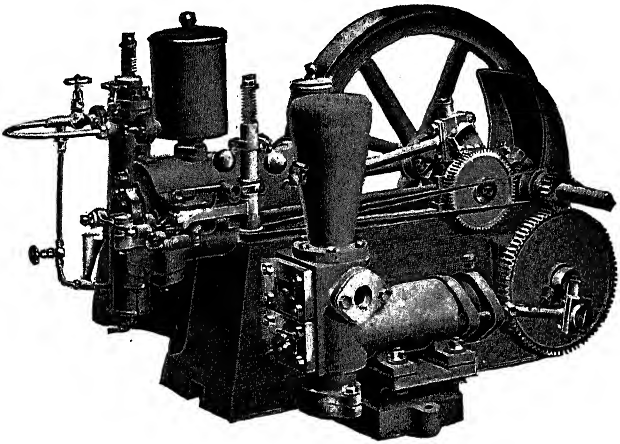


FIG. 62.

town or village water-supply. This outfit consists of 13 H. P. oil engine connected by friction-clutch to the shaft of a triplex pump having cylinders $6\frac{1}{2}''$ diameter and 8″ stroke.

The amount of water delivered by this outfit is approximately 165 gallons per minute, with total average lift of 195 ft. The cost of fuel for running is

about 13 cents per hour. Practically, no attention is required beyond starting the engine and occasional lubrication.

Fig. 62 shows a small outfit suitable for supplying water to a country-house, and consists of $1\frac{1}{2}$ H. P. engine and pump capable of delivering 1200 gallons of water with 150 ft. total lift.

To calculate the theoretical H. P. required to raise a

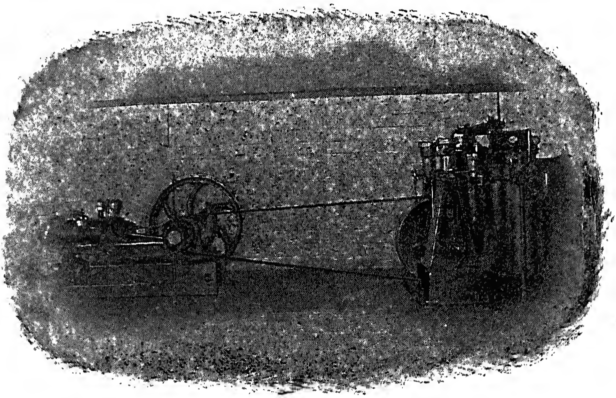


FIG. 63.

given amount of water, multiply the number of gallons to be delivered per minute by 8.3, which gives the weight; again, multiply by the total required lift in feet, and divide the result by 33,000, thus:

$$\text{H. P.} = \frac{\text{Number of gallons} \times 8.3 \times \text{height of lift}}{33{,}000}$$

Example: 165 gallons 195 feet lift

$$\frac{165 \times 8.3 \times 195}{33,000}$$

= 8 H. P. actually required to lift water.

The friction of the moving parts of the pump has to be overcome, and for this and other losses allowance is usually made by figuring the efficiency of the pump (in the smaller size) at 60 per cent. to 70 per cent.

Oil Engines Driving Ice and Refrigerating Machines.

Oil engines are now being used in connection with small ice and refrigerating machines.

Fig. 63 represents a plant of this description, consisting of an oil engine belted direct to a refrigerating machine used in this instance for cooling a butcher's cold-storage box.

The refrigerating machines are rated according to the amount of ice they are assumed to displace. A one-ton machine is one which will effect the same cooling in twenty-four hours which a ton of ice would do in melting. The chief advantage of the refrigerating machine is that while the ice can only produce a temperature of 35° Fahr. and upward, the refrigerating machine can be operated to produce any temperature which may be desired.

In the process of refrigeration, the work which the

oil engine has to do is to drive a compressor, and therefore the same principles may be applied to this machine as to the ordinary air-compressor already discussed. We need only to know how much gas has to be compressed and the conditions upon which to base the calculation for the work done in the compressor. It is the practice of refrigerating-machine makers to allow about 4.5 cubic ft. displacement per ton of refrigeration—that is to say, a 10-ton machine is one having capacity of pumping 45 cubic ft. of gas per minute.

In the case of the ordinary compressor, we have only to consider the final pressure, since the initial pressure is always that of the atmosphere. In the case of the refrigerating machine, however, this is not the case, for the gas being circulated in a closed circuit may have not only a varying final pressure, but also a varying suction pressure. These pressures depend upon the temperatures obtaining in the cold room and in the condenser in a manner which it is not necessary to consider in detail. The initial pressure and the final pressure being known, the mean pressure may be calculated in the ordinary way.

To facilitate this calculation, table No. IV. may be consulted. The vertical left-hand column gives the initial pressure corresponding to the temperatures named in the second column, these being the temperatures *inside* the cooling pipes. The top horizontal line gives the pressure corresponding to the temperatures in the second horizontal line. These temperatures are those obtaining in the condenser.

TABLE IV.—MEAN PRESSURE OF DIAGRAM OF GAS (AMMONIA) COMPRESSOR.

Refrigerator Pressure and Temperature.		Condenser Pressure and Temperature.								
		103	115	127	139	153	168	184	200	218
		65°	70°	75°	80°	85°	90°	95°	100°	105°
4	— 20°	41.46	43.91	46.34	48.77	51.23	53.68	56.11	58.54	60.99
6	— 15°	42.72	45.38	47.90	50.74	53.40	56.08	58.86	61.40	64.08
9	— 10°	44.40	47.38	50.33	53.29	56.25	59 20	62.16	65.14	68.09
13	— 5°	45.86	49.15	52.42	55.70	58.97	62.25	65.53	68.81	72.08
16	0°	46.94	50.56	54.16	57.78	61.40	65.00	68.62	72.22	75.84
20	5°	47.74	51.73	55.70	59.68	63.67	67.66	71.62	75.61	79.61
24	10°	48.04	52.40	56.77	61.13	65.51	69.86	74.24	78.59	82.97
28	15°	47.88	52.67	57.44	62.23	67.02	71.81	76.60	81.39	86.18
33	20°	47.08	52.30	57.53	62.75	67.98	73.23	78.46	83.68	88.91
39	25°	45.06	51.34	57.05	62.75	68.46	74.17	79.88	85.58	91.29
45	30°	43.16	49.71	55.92	62.14	68.35	74.56	80.77	86.98	93.19
51	35°	40.52	47.26	54.02	60.76	67.52	74.28	81.02	87.78	94.52

The mean pressure corresponding to any two known conditions may therefore be taken from the table; for example, with a suction pressure of 28 and a condenser pressure of 153, the mean pressure is 67.02 pounds. The work required to produce a ton of refrigeration, therefore, would be

$$\text{H. P.} = \frac{PLAN}{33{,}000},$$

in which

$P = 67.02$ pounds.
$L = 4.5$ feet.
$A = 144$ square inches $= 1$ sq. ft.
$N = 1$.

Substituting these values, the horse-power is 1.32. No allowance is here made for friction, and in small refrigerating machines this should be extremely liberal.

Moreover, on reference to the table it will be seen that the machine may happen to be called upon to work under conditions where the mean pressure will be very much increased; such, for example, when the back pressure is 51 lbs. and the high pressure is 218 lbs. Under these circumstances the mean pressure will be 94.52 instead of 67.02. For these reasons it is not safe to provide for a refrigerating machine of small dimensions a power much less than about 3 H. P. per ton of refrigeration. Under ordinary conditions of running, less than this, and frequently only one-half of this will be required, but provision should be made for taking care of extreme conditions.

FRICTION-CLUTCHES.—Where engines of 10 H. P. or over are installed, it is a great advantage to have a friction-clutch pulley added. This can be attached either to the engine crank-shaft or to the intermediate or main shaft. Fast-and-loose pulleys are sometimes substituted for the friction-clutch.

With either friction-clutch or fast-and-loose pulleys the advantages gained are, first, the ease with which the engine can be started, the loose or friction-clutch pulley only instead of the whole shaft has to be turned when the plant is started, and, secondly, in case of accident or other emergency necessitating the quick cessation of the revolving machinery, this can be accomplished at once by simply moving over the handle of the friction-clutch and pulley. Otherwise without the clutch the heavy fly-wheels of the engine remain revolving for a minute or so after the fuel of the engine is turned off, and being directly connected by belt to the shafting and machinery, the whole plant is in motion while the momentum of the fly-wheels exists.

Friction-clutches are made of various designs by several manufacturers. That shown in Fig. 63*a* is especially adapted for explosive engines. It consists of a carrier which bolts to the regular bosses on the fly-wheel of the engine, this carrier acting as the journal of the pulley, and the mechanism of the clutch is enclosed in the same. The clutch has a side grip. The pulley, otherwise loose, is thrown into connection with the engine fly-wheel by simply pushing in a spindle on which a hand-wheel revolves loosely. Two rollers are mounted on the end of the spindle, and bearing on

these rollers are the levers which in turn are pivoted to the gripping plate and a lug on the levers abuts against the adjusting screw. The inward movement of the spindle forces these levers apart and draws the gripping plate in, thus gripping the pulley in a circular vise

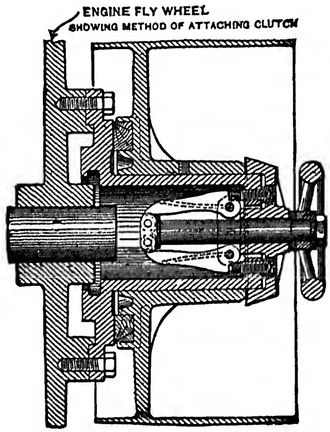


FIG. 63*a*.

between the flange on the carrier and the gripping plate. To release the clutch the spindle is pulled out, and thereby the strain on the levers is removed, thus allowing the pulley to run loose. This clutch is known as the B and C Friction Clutch Pulley.

CHAPTER VII.

INSTRUCTIONS FOR RUNNING OIL ENGINES.

THE attendant, in order to obtain the best results from an engine, should first fully understand the principle by which the engine he is running works and the conditions which it is essential should exist in the cylinder to procure proper explosion and combustion. These conditions are practically the same in all types of oil engines. The explosive mixture consists of hydrocarbon gas and atmospheric air, the gas being formed from kerosene oil previously gasefied or vaporized and properly mixed with air by one or other of the different methods, as described in Chapter I. This mixture is then compressed by the inward stroke of the piston before ignition with the two-cycle type of engine. The mixture is afterward ignited by hot tube, electricity, heated surfaces, or otherwise, as also described in Chapter I., and the required impulse is then obtained at the piston. If for any reason these conditions are not existing, proper explosion and combustion will not follow. The several reasons which prevent proper explosions being obtained are very fully described in Chapter III. on "Testing."

The conditions necessary to insure proper working are as follows:

(*a*) Oil supply to the vaporizer or combustion chamber delivered at the correct time, and in such quantity as to form proper explosive mixture. Efficient supply of air.

(*b*) Sufficient pressure in the cylinder by compression before ignition.

(*c*) Correct ignition of the gases, the ignition taking place at the proper time.

CYLINDER LUBRICATING OIL.—It is essential that a suitable lubricating oil be used for the piston. The great heat evolved in the cylinders of explosive engines renders this essential.

The lubricating oil recommended for this purpose is a light mineral oil having a flash point of not less than 360° Fahr. and fire test 420° Fahr. Gravity test 25.8, and having a viscosity of 175 (Saybold test). If waste-oil filter is used, the oil filtered must not be employed for lubricating the piston at any time.

The following are instructions as formulated by the makers of the different engines, each of the four types of vaporizers being here represented, as well as the different kinds of igniting devices.

HORNSBY-AKROYD TYPE.

The method of working is explained in Chapter IX., giving general description of these engines. The oil-tank in the base of the engine should be filled

and the oil pumped up by hand until it passes the overflow pipe. The water-tanks if used must also be filled to the top and the cylinder water-jacket also be full of water before starting.

PREPARING TO START THE ENGINE.—On those engines in which the vaporizer is partially water-jacketed, the valve on the inlet water-pipe should be closed before commencing to heat the vaporizer for starting, and opened, or partially opened, when running.

TO HEAT THE VAPORIZER.—A coil lamp is used (see illustration, Fig. 64) for this purpose; the lamp reservoir should be nearly filled with oil. A little kerosene should then be poured into the cup containing asbestos wick under the coil and lighted. When this has nearly burnt out, pump up the reservoir with air by the air-pump, when oil vapor will issue from the small nipple, and on being lighted will give a clear flame. When it is required to stop the lamp, turn the little thumb-screw on the reservoir-filling nozzle and let the air out, and remove the lamp from the bracket. The nipple at any time can be cleaned with the small prickers which are supplied for this purpose. Should the U-tubes get choked up, the lower one can be gotten at by unscrewing the joint just below it, and the other one by screwing out the nipple from which the oil vapor issues. The heating of the vaporizer is one of the most important duties to be attended to, and care must be taken that it is made hot enough before starting. The attendant must see that the lamp is burning properly for five or ten minutes, or sometimes a little longer, according to the size of the engine. If, however, the

lamp is burning badly, it may take longer to get the proper heat. It is most important that the lamp should be carefully attended to.

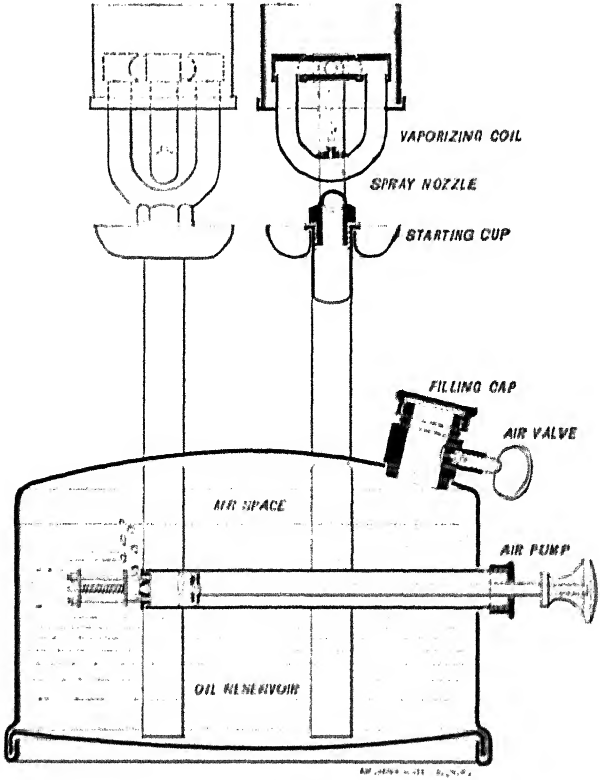


FIG. 64.

TO START THE ENGINE.—Place the starting handle to position "Shut," and work the pump-lever up and down until the oil is seen to pass the overflow-valve.

Then turn the handle to position "Open," work the pump-lever up and down again, one or two strokes, then give the fly-wheel one or two turns, and the engine will start readily. There is also a handle upon the cam-shaft, which, when starting the engine, must be placed in the position marked "To Start," and immediately the engine has gotten up speed this handle should be placed in position marked "To Work."

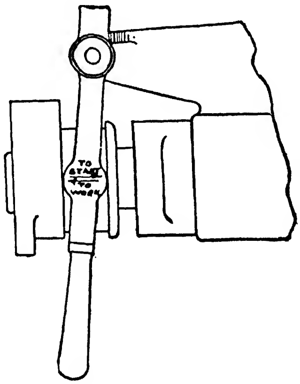


FIG. 65.

(See Fig. 65.) When it is required to stop the engine, turn the starting handle to the position marked "Shut." If too much oil is pumped into vaporizer before starting it will be difficult to start up.

OILING ENGINE.—See that the oil-cups on the main crank-shaft bearings are fitted with proper wicks and with other oil-cups are filled with oil. Oil the

small end of the connecting-rod which is inside the piston, also the bearings on horizontal shaft and the skew-gearing, the rollers at the ends of the valve-levers and their pins, and the pins on which the levers rock, the governor spindle and joints, the bevel-wheels which drive same, and the joints that connect the governor

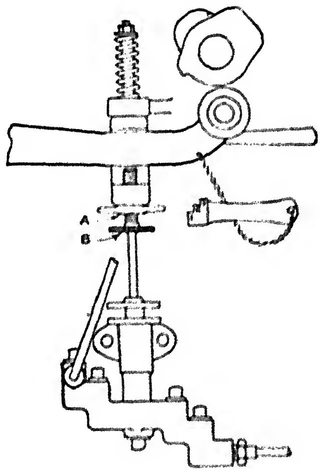


FIG. 66.

to the small relief-valve on the vaporizer valve-box. For such purposes, none but the best engine oil should be used.

OIL-PUMP.—When the engine is working at its full power the distance between the two round flanges *A* and *B* on the pump-plunger should be such that the gauge "1" will just fit in between the flanges. (See

Fig. 66.) The other lengths on the hand-gauge marked "2" and "3" are useful for adjusting the pump to economize oil when running on a medium or a light load. Do not screw down the pump packing tight enough to interfere with the free working of the plunger.

RUNNING ENGINES LIGHT OR NEARLY SO.—When engines are required to run with light or no load, it is best to alter the stroke of the pump to supply only sufficient oil to keep the engine running at full speed, so that the governor occasionally reduces the oil. The inlet water-pipe to the vaporizer-jacket should be closed when running light also.

AIR-INLET AND EXHAUST VALVES.—See that the air-inlet and exhaust valves are working properly and drop onto their seats. They can at any time, if required, be made tight by grinding in with a little flour of emery and water. The set-screws at the ends of the levers that open these valves must not be screwed up so high that the valves cannot close; this can be ascertained by seeing that the rollers at the other end of the levers are just clear of the cams when the projecting part of the cams is not touching them. (See Fig. 67.)

VAPORIZER VALVE-BOX.—In this box there are two valves. The vertical one is regulated by the governor, and when the engine runs too fast the governor pushes it down, thus opening it and allowing some oil to overflow into the by-pass, which should only allow oil to pass when the governor presses it down, or when the starting handle is turned to "Shut." The horizontal

valve in this box is a back-pressure valve, and should a leakage occur it may be discovered by slightly opening the overflow-valve (by pressing it down with the hand), when, if there is a leakage, vapor will issue from the overflow-pipe, and in that case the valve should be examined, and, if necessary, be taken out for inspection and ground on its seat with a little emery flour and water. If the horizontal valve and sleeve are taken out, care should be taken, in replacing them, to use the same thickness of jointing material as before.

OIL-PIPES.—The pipe from the pump to the vaporizer valve-box has a gradual rise from the pump; if

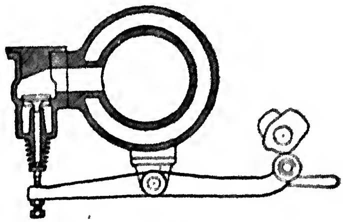


FIG. 67.

otherwise, an air-pocket would be formed in which air would be compressed upon each stroke of the pump, and thus allow the oil to enter slowly and not as it should do, suddenly. If the oil gets below the filter at any time, work the pump by hand a few minutes, holding open the overflow-valve in the vaporizing valve-box, so as to get the air well out of the pipes. The oil-filter should be taken out and cleaned occasionally.

Spray Holes.—It may be desirable to take off the vaporizer valve-box and clean the little hole or holes through which the oil issues. The reamers, or small wires supplied, are not for increasing the size of the hole, but are simply for cleaning it at any time.

Testing Oil-Pump.—See that the pump gets its proper oil supply. Disconnect the oil-supply pipe union attached to vaporizer valve-box, and give the

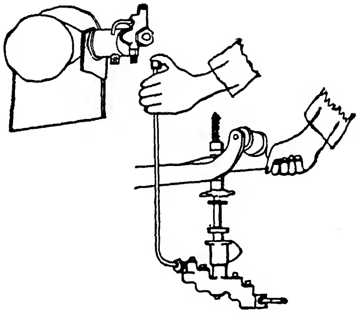


Fig. 68.

pump two or three strokes so as to pump oil up; then press the thumb firmly on the end of the pipe, as shown in illustration, Fig. 68. Pump both by a sudden jerk, and afterward by a steady pressure. If the plunger yields to a sudden jerk and no oil has gotten past the thumb over the top of the delivery-pipe, then the pump or the pipes contain air. If the plunger does not yield to a sudden jerk, but slowly falls under a constant pressure, then the suction-valves of pump are

not tight. If necessary, the valve-seats can be renewed by lightly driving the cast-steel ball valves onto their seats with a small copper punch. If it is required to see that the vaporizer valve-box is in order, take off the vaporizer valve-box body and sleeve, and connect them to the oil-supply pipe from the pump, so that the jet from the spraying hole can be directed where it can be seen. Work the pump by hand, when the jet produced should be clear, with distinct and abrupt pauses between each delivery.

THE GOVERNOR "HUNTING."—This may be caused by the joints or spindle of the governor becoming bent, dirty, or sticky, and requiring cleaning. If the pump is not giving a regular supply of oil, it may sometimes cause the governor to hunt, and the engine would run irregularly. This may occur when the engine is first started.

THE CROSSLEY PATENT TYPE.

STARTING.—Heat the ignition-tube by means of the lamp in the usual way. The pressure (about 60 lbs.) necessary to raise the oil to the lamp in this engine is taken from the oil-tank, the air pressure before starting being created by hand. This lamp heats both the ignition-tube to a good red heat and vaporizer blocks to less heat simultaneously. The necessary pressure to raise the oil to the lamp is maintained by the pump actuated from the cam-shaft when the engine is running.

PRIMING CUP.—Fill the little brass priming cup or

of the vaporizer cover with oil; open the valve the oil pass through into the vaporizer, and it it again. Leave the wire on the chain out of surer. Place the exhaust roller over to engage e one-half compression cam; turn the fly-wheel e crank-pin is about one inch above the hori- both valves being closed); open the stop-valve nd of air-receiver; connect up the oil-pump by g the back-pin, having first made a few strokes : hand-pump until the oil-pipe is full up to the r, and turn the quadrant on air-throttle valve. ine is now ready to start, and the air under pres- n receiver may be let in. Loosen the screw of alve; open the valve by means of the loose lever, l open until the crank has just passed the verti- ion. This impulse will be sufficient to turn the l a few times, during which the piston will re- ular impulses. The exhaust roller may then be ff the one-half compression, when full speed steadily attained.

on as convenient the screw on the starting ly be unscrewed to allow the receiver to be- harged again. Should the engine miss explo- d fail to attain full speed, then turn the lid of partly around and give a little extra supply m a hand-can.

UPPLY.—At full speed the air-throttle must be o admit more air, and the amount must be s to whether the engine ignites its charges or much air will cause it to miss fire—too little s too sharp firing. If the receiver is not

not tight. If necessary, the valve-seats can be renewed by lightly driving the cast-steel ball valves onto their seats with a small copper punch. If it is required to see that the vaporizer valve-box is in order, take off the vaporizer valve-box body and sleeve, and connect them to the oil-supply pipe from the pump, so that the jet from the spraying hole can be directed where it can be seen. Work the pump by hand, when the jet produced should be clear, with distinct and abrupt pauses between each delivery.

THE GOVERNOR "HUNTING."—This may be caused by the joints or spindle of the governor becoming bent, dirty, or sticky, and requiring cleaning. If the pump is not giving a regular supply of oil, it may sometimes cause the governor to hunt, and the engine would run irregularly. This may occur when the engine is first started.

THE CROSSLEY PATENT TYPE.

STARTING.—Heat the ignition-tube by means of the lamp in the usual way. The pressure (about 60 lbs.) necessary to raise the oil to the lamp in this engine is taken from the oil-tank, the air pressure before starting being created by hand. This lamp heats both the ignition-tube to a good red heat and vaporizer blocks to less heat simultaneously. The necessary pressure to raise the oil to the lamp is maintained by the pump actuated from the cam-shaft when the engine is running.

PRIMING CUP.—Fill the little brass priming cup or

the top of the vaporizer cover with oil; open the valve and let the oil pass through into the vaporizer, and then shut it again. Leave the wire on the chain out of the measurer. Place the exhaust roller over to engage with the one-half compression cam; turn the fly-wheel until the crank-pin is about one inch above the horizontal (both valves being closed); open the stop-valve on the end of air-receiver; connect up the oil-pump by replacing the back-pin, having first made a few strokes with the hand-pump until the oil-pipe is full up to the measurer, and turn the quadrant on air-throttle valve. The engine is now ready to start, and the air under pressure from receiver may be let in. Loosen the screw of starter valve; open the valve by means of the loose lever, and hold open until the crank has just passed the vertical position. This impulse will be sufficient to turn the fly-wheel a few times, during which the piston will receive regular impulses. The exhaust roller may then be moved off the one-half compression, when full speed will be steadily attained.

As soon as convenient the screw on the starting valve may be unscrewed to allow the receiver to become recharged again. Should the engine miss explosions and fail to attain full speed, then turn the lid of measurer partly around and give a little extra supply of oil from a hand-can.

Air Supply.—At full speed the air-throttle must be opened to admit more air, and the amount must be judged as to whether the engine ignites its charges or not; too much air will cause it to miss fire—too little air causes too sharp firing. If the receiver is not

charged, and it is required to start engine by hand, pull around the fly-wheel and get up as much speed as possible before putting the governor blade in position for engaging with the governor mechanism which opens the gas-valve.

VAPORIZER BLOCK.—The vaporizer block must be well heated previous to starting; otherwise unvaporized oil will be carried over into cylinder, and thus make starting uncertain until the oil has all passed away in evaporation. This may also cause puffs of vapor to rush out of the air inlet at the top of the chimney, preceded by a slight explosion in the vaporizer block. This is caused by late ignition in cylinder, and is due to insufficient vaporization or to the ignition-tube not being hot enough.

VAPOR VALVE.—If small puffs of vapor issues out of the air-pipe of the chimney every other revolution while the engine is running, it is a proof that the vapor-valve is not tight and must be cleaned and ground on its seating.

CAMPBELL OIL ENGINE.

STARTING.—Before starting the engine, see that the vaporizer is thoroughly well heated. The lamp under the vaporizer should burn with a long, bright flame. When the vaporizer is sufficiently heated, throw the governor drop-lever down, thus holding the exhaust-valve open and relieving the compression. While this lever is held down, give a quarter or a half turn of the

oil-cock; then turn the fly-wheel quickly four or five revolutions, and allow the governor drop-lever to be free. It will swing up clear of the exhaust-lever and allow a charge of air and oil to be driven into the vaporizer; the engine should then commence working. After the engine has started, turn on a little more oil. If the oil taken into the vaporizer should not explode properly, the oil-cock must be shut and opened again quickly to allow any superfluous oil which has lodged in the vaporizer to be drawn out of it and vaporized. When using a heavy oil, open the inlet-valve to allow more air to flow into the vaporizer.

AIR AND OIL SUPPLY.—Too much oil passing to the vaporizer will cause the engine to miss exploding or to explode irregularly. To increase the air supply, slacken the nuts and tension of air-inlet valve; by tightening the nuts and spring, the air supply is decreased.

IGNITION-TUBE.—See that the inside of the ignition-tube is kept clear from oil, and keep all the valves clean and the governors free from oil and dirt. When the engine is running properly, the quantity of oil required is the same, whether the engine is running at light or heavy load.

GOVERNORS.—The governors cut out some of the charges at light loads and admit more charges of oil at heavy loads; each charge, however, has the same composition of vapor and air.

THE PRIESTMAN TYPE.

STARTING.—Open the drain-cock in the vaporizer and see that the vaporizer contains no oil; then close the cock. Fill the oil-tank to the small upper-pet cock, through the strainer provided and screw down the relief air-valve. Lubricate the piston wrist-pin and the crank-bearing between the fly-wheels. Drop a little oil on the pump-piston and in the oil holes of the bearings of the large gear-wheels, the eccentric, and all other bearings. Mineral oil must not be used on the governor oil spindle which projects into the spray-maker.

ELECTRIC IGNITER.—Raise the electric fork-handle slightly. This is done in order to produce the igniting spark somewhat later for starting than is required when the engine is running at full speed. Turn the fly-wheels forward until the small knob on the cam-shaft has just passed the contact with the forks, and the crank-pin is then just clear of the large gear-wheel.

HEATING VAPORIZER.—Heat the vaporizer until the lower part of the feed-pipe leading to the inlet-valve is too hot to be comfortably held by hand. When the vaporizer is sufficiently heated, pump up 6 or 8 lbs. gauge air pressure in the oil-tank with the hand-pump; open the oil-cock, and then give the fly-wheels a few turns with the starting handle. After starting, move the electric fork-handle down as far as it will go.

AIR SUPPLY.—Set the air-relief valves for giving about 8 to 10 lbs. air pressure in the oil-tank. The most suitable running pressure in a given locality as indi-

cated by the gauge, has to be determined by experiment. With the air pressure too low or too high, the engine may miss explosions. The best test for this is the color of the ignition-plug. When the pressure is right, the plug will be perfectly clean. If the plug is coated with an oily black substance, it is a sign of too much oil—that is, too high a pressure. To stop the engine, turn off the oil-cock. When stopped, see that the electric circuit is not closed, or the battery energy will be wasted.

General Remarks.—If an oil engine is working properly and efficiently, it should run smoothly to the eye, without knocking either in the cylinder or bearings. The piston should continue to work clean and be well lubricated, without any apparent carbon or gummy deposit. The exhaust gases at the outlet-pipe should be invisible or nearly so. The explosion should be regular and should be only reduced in pressure when the governor is reducing the explosive charge and allowing only part or none of the charge of oil to enter the cylinder.

If the piston is black and gummy, or if the exhaust gases are like smoke, then the combustion inside the cylinder is recognized as being incomplete, and the cause should at once be ascertained and remedied.

Bad combustion may be due to several reasons, but is chiefly caused by improper mixture of air and gases in the cylinder, due either to too much oil entering into the vaporizer or to insufficient amount of air being drawn in mixed with the hydrocarbon gas. To remedy this defect, examine the oil-inlet valves or spraying de-

vice carefully; also see that air and exhaust valves are allowed to drop freely on their seats, and that springs or other mechanism for closing the valves are in good shape. Examine piston-rings and ascertain that the rings are in good order and are not allowing leakage of air to pass them.

REGULATION OF SPEED.—To alter speed of the engine with the hit-and-miss type of governor, the spring is strengthened or the weight reduced to increase speed. The weight is effectively increased by moving it toward the end of the lever away from the fulcrum-pin, and *vice versa* to reduce speed. The strength of the spring is increased by tightening down the thumb-screw nut. With the Porter type of governor where counterbalance with movable counterweight is provided, the speed is accelerated by increasing the supplementary weight, or by placing it nearer the end of the lever. If the centrifugal force of the revolving weights is controlled by a spring instead of weight, then the speed is increased by strengthening the spring.

REVERSING DIRECTION OF ROTATION.—In order to reverse the direction of rotation of an explosive engine, it is necessary to change the relative position of the cams actuating the air and exhaust valves and fuel supply so as to alter the periods of opening and closing of these valves, and also to change the period of fuel supply. In those engines in which one cam controls both the air-inlet valve and the fuel supply, the shifting of this one cam alone effects the change necessary.* Where the fuel supply is operated separately, the cam

*The position of the exhaust cam to conform to the diagrams in Fig. 69 is changed by alteration of the gearing in the cam shaft.

or eccentric controlling this mechanism must be moved correspondingly with the air-valve cam.

The following diagrams give the correct positions

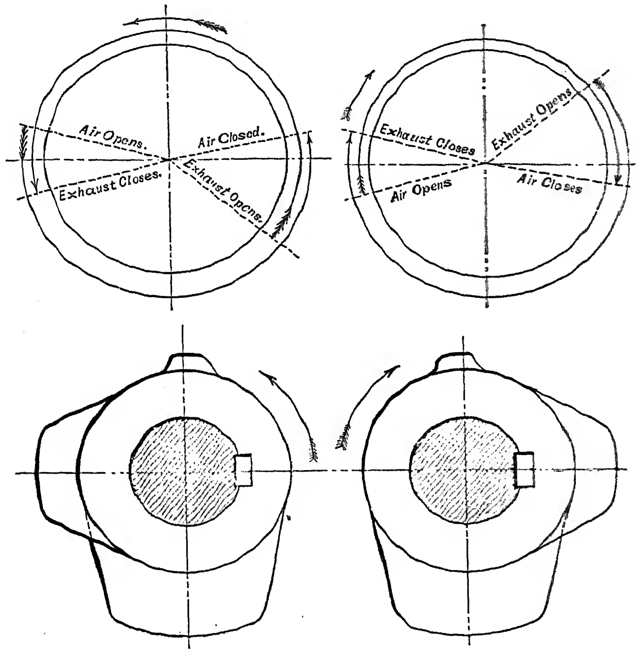


FIG. 69.

of the opening and closing of the valves when the engine is running in each direction, and the cams as set for each case are shown in Fig. 69, the slot for key-way in the air-inlet cam having been changed only.

Where the air-inlet valve is automatic and the exhaust valve only is actuated from the crank-shaft, then, to reverse the direction of rotation of the crank-shaft, the position of the exhaust-cam only is changed, corresponding to the position as marked for the exhaust valve in diagram shown in Fig. 69.

The lip for regulating the compression when starting the engine only, which is usually found on the exhaust cam, will require adjustment when the engine is reversed so as to close the exhaust valve when approximately one-half the compression stroke has been completed. The direction of rotation for which the cams of the engine are adjusted can be ascertained by turning the fly-wheel until the exhaust cam commences to open the exhaust valve. If the exhaust valve is opened when the crank-pin is above the outward centre, as shown on the diagram to the right in Fig. 69, then the direction of the engine is "over" or away from the cylinder. When the exhaust valve opens below the centre of the crank-pin, as shown in diagram to the left in Fig. 69, then the direction of rotation of the fly-wheel will be "under"; that is, the upper part of the fly-wheel will revolve toward the cylinder.

CHAPTER VIII.

REPAIRS.

OIL ENGINES as made by most of the makers are of substantial construction, with ample bearing surfaces, and consequently require few repairs. The lower initial pressures of explosion evolved in oil engines as compared with some gas and gasoline engines considerably lessens the severe shock to the piston and to the crank-shaft bearings and connecting-rod bearings. All machinery requires repairs more or less according to the care that it receives, and oil engines are not an exception to this rule.

THE PISTON should be drawn out occasionally; this is done by uncoupling the connecting-rod crank end bearings and pulling the piston out. Chain-block is sometimes added to the installation of large engines, and it is a very useful adjunct when it is required to take out the piston or when other repairs have to be made. Where no arrangement of this kind is available when the piston is to be taken out, wooden packing is placed in the engine-bed, on which the piston can rest as it is drawn out. Care should be taken that the weight of the piston as it is drawn from the cylinder does not fall on the piston-rings or they may thus be broken.

With the vertical type of engine the piston is taken out from the top, the cylinder head and other parts having been removed.

The piston should be washed with kerosene and well cleaned. When putting piston back in place, each ring should be put separately in exact position in its groove as regards the dowel-pin in piston groove before the ring enters the cylinder. The piston, the rings, and the inside of the cylinder must all be carefully cleaned and well lubricated with proper oil before being again put in place. Where the rings require cleaning, this can be accomplished by washing with kerosene. If, however, the piston-rings are to be taken off the piston, they must be separately sprung open by having pieces of sheet metal about 1-16″ thick and about ½″ wide inserted between ring and body of piston.

Air and exhaust valves should also be periodically taken out, cleaned and examined, and, if necessary, reground in. Powdered emery or glass powder is considered satisfactory to grind the valves in with.

Care should be taken, in replacing valves, that they are clean and free from rust or carbon, and are allowed to drop on their seats freely and do not stick in their guides.

The crank-shaft bearings will periodically require taking up as they show signs of wear and commence to knock or pound. Usually, for this adjustment, liners are left between the cap and the lower half of bearings. These liners can be occasionally reduced in thickness, so that the cap is allowed to come down close on to the shaft. Great care must be taken, in

tightening down the bearing again after adjustment, that it is not bolted down too tight on the shaft bearings; otherwise heating will result and the bearings and journal may be cut and damaged in running.

The connecting-rod bearings will require adjustment more often than the crank-shaft or main bearings.

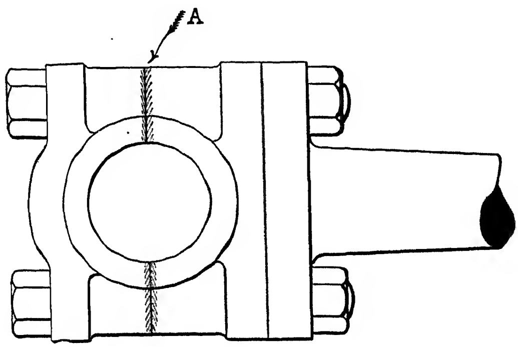


FIG. 70.

When this is necessary, the engine will be heard to knock at each revolution, and then the bearing should be taken apart at the crank-pin bearing and about 1-64″ filed off. (See *A*, Fig. 70.) As with the crank-shaft bearings, great care, in putting bearing back in place, must be exercised, first to see that it is thoroughly clean and free from dirt, and also, when readjusted, that it has a slight motion sideways and can thus be moved by hand.

When fitting new piston-ring, it is well to place the

ring in the cylinder correctly; it should have slight space, about 1-64″ left for the expansion between the joint which will take place when heated in working.

After fitting new worm or spur gearing to the valve motion, the positions of the cams should be tested by turning the fly-wheel over by hand. The correct positions of the cams are shown on diagram, Fig. 32.

The oil-filter requires occasional renewing; this can be made of muslin placed between wire gauze, as shown in Fig. 28. The oil-supply pump-valves, if they consist of steel balls, can be refitted to their seats by being tapped when in place with copper plug or piece of wood. When renewing governor parts, care must be taken that the new part is free and works without friction; this is very essential where close regulation of speed is required.

CHAPTER IX.

OIL ENGINE TROUBLES.

THE requirements for proper working of the oil engine have been already mentioned in Chapter VII. as follows: Proper oil and air supply to the cylinder or vaporizer, proper mixture or combination of air and vapor, correct and properly timed ignition. Defects which may cause improper working have also been referred to in Chapter III. on testing.

The following remarks are chiefly applicable to the operator, and refer to difficulties which may possibly be encountered in the actual use of the oil engine.

TROUBLES OF IGNITION.

THE ELECTRIC IGNITER.—This igniter is described in Chapter I. Failure in operation is generally due to the following causes:

BREAKAGE IN ONE OR OTHER OF THE ELECTRICAL CONNECTIONS.—To discover the breakage test with a length of wire in the hands bridged across between the terminals of the connection which is thought to be defective, the circuit through the cam-shaft being closed. If a spark is then given off the defect has been located and a new connection should be put in place. In

some instances a spark is not produced because the battery is run down; this defect can be ascertained by testing the battery with a small volt meter or by bringing both terminals in contact one with another from the battery; a strong spark should then be seen. If the battery is run down, it must, of course, be recharged or renewed. The terminals in the cylinder must always be clean and free from carbon deposit. This is important especially with a jump-spark plug igniter, as the terminals in the cylinder will sometimes become carbonized or corroded, thus forming a path for the current to flow across without causing any spark.

Failure to obtain electric spark ignition may occur from bad insulation of the plug. In this case a new plug should be substituted for the defective one. In some instances the electric spark is not procured because the plug is short-circuited, due to moisture. To overcome this the plug must be thoroughly cleaned and dried out or a new plug must be substituted. With the type of igniter having movable electrode, owing to friction or carbonizing, the two electrodes may be prevented from touching. In this case the moving electrode should be eased or cleansed and allowed to come freely in contact with each other.

The timing of the ignition with the electric igniter is regulated by altering the time of contact. The period of ignition varies according to the speed of the engine. With a high speed the ignition should take place just before the crank-pin arrives at the dead centre; with a slow-speed engine the time of ignition can be slightly later; that is, the ignition may take place as the crank-

pin passes the dead centre. When starting the engine, the ignition is retarded until the normal speed of the engine is attained.

TUBE IGNITER.—Troubles with this form of igniter are generally due to corrosion internally of the tube. This is remedied by taking the tube out and thoroughly cleaning it. In other instances ignition is not obtained because the tube is not properly heated. The temperature of the tube should be maintained at a good red heat. With the tube igniter it is essential that the gases can properly enter it. The timing of ignition with this form of igniter can be varied by changing the length of the tube or by altering the part of the tube which is heated. If an earlier ignition is required, the tube should be heated nearer to the cylinder end, or a shorter tube should be used. If it is required to retard the time of ignition, the tube can be heated further from the cylinder, and accordingly the gases to be ignited will not come in contact with the heated part so rapidly.

AUTOMATIC IGNITER.—In order to procure proper ignition with this form of igniter, it is essential that the compression of the air and gases is efficient. This pressure varies in different types of engines, and, as will be seen from the indicator cards shown in Chapters III. and X., is from 50 to 70 lbs. The mixture of air and oil vapor must also be correct. Failure to obtain an ignition with this type of engine is usually due to too much oil having been allowed to enter the vaporizer or cylinder, or to the fact that no oil at all has entered the vaporizer, or, as already stated, to fail-

ure to obtain proper compression. Ignition, of course, cannot be obtained when starting unless the vaporizing chamber or retort has been properly heated.

Oil Supply.—If the oil supply is defective, the fault can be ascertained by careful examination. Disconnect the oil-supply pipe and see that oil flows freely from the tank. Sometimes the oil filter in the tank will become clogged and will not allow the oil to flow through it. If oil is supplied by a pump, then test the pump, as shown on page 147. Failure of the pump to operate properly is due to leaky valves or to the packing around the plunger, allowing air to leak by, and thus the proper pressure in the pump is lost.

The oil supply may fail by reason of leakage in the oil pipes. This may easily happen where the oil tank is placed below the level of the engine and the oil has to be raised from the tank by pump. In such a case the engine may operate when the pump is working at full stroke, whereas otherwise no oil will be delivered to the cylinder or vaporizer.

Air Supply.—Defective air supply is due to leakage in the piston-rings, piston, or to leakage in the air and exhaust valves. The compression in the cylinder is, of course, governed by the air supply, and a leakage in the valves or piston can be tested by simply turning the engine backwards. With proper compression it should be difficult to turn the crank-pin past the inward dead centre during the compression period.

Knocking.—An engine working properly should be quiet in operation. Knocking may be due to either loose bearings in the connecting-rod, piston or crank-

pin end, to loose fly-wheel keys, or to improper timing of ignition. The first two defects can be ascertained by examination. The timing of ignition is most easily ascertained from the indicator card. (See page 76.)

Loss of Power.—This may be due to increased friction in the engine, which friction may be caused by bad lubrication of the piston or the piston becoming gummed up, due to improper combustion or to the use of improper lubricating oil. (See page 140.) Loss of power may also be due to heated bearings. Either of these causes can be easily ascertained. Insufficient oil or fuel supply due to the wearing of the moving parts and consequent reduction of the pressure of explosion is sometimes responsible for the loss of power. To overcome this the supply of fuel can be slightly increased. That the proper amount of fuel is being supplied can be roughly ascertained by the color of the exhaust gases. If too much oil is supplied the exhaust gases will be plainly visible. With the correct oil supply the exhaust gases will be invisible or nearly so.

Piston Blowing.—This is due to the various following causes: Improper lubrication, to the piston-rings leaking, to the piston-rings having become clogged, or to the cylinder having become cut or worn. It is also sometimes due to over-expansion of the cylinder, caused by over-heating and insufficient water supply. If the blowing of the piston cannot be remedied by proper lubrication or by thoroughly cleaning the piston-rings new piston-rings must be put in place. In some cases it is even necessary to re-bore the

cylinder and have new piston and rings. The blowing of the piston may be also caused by improper combustion due to too great an oil supply or insufficient air supply. Escape of vapor from the open end of the piston, which is thought to be a leakage, is sometimes caused by the splashing of the oil on the overheated bearings or the heated portion of the piston. This can be ascertained by stopping the engine. If vapor continues to escape when the engine is at rest, its cause is apparent, and then the supply of lubricating oil to the cylinder can be reduced.

Explosions in the Muffler or Silencer.—A loud report may sometimes be heard, caused by the explosion in the exhaust pipe or muffler. This is due to the gases passing through the cylinder unconsumed and then becoming ignited in the silencer. It is not possible to create a dangerous pressure in this way, and as the silencer is usually a heavy cast-iron chamber and always open to the atmosphere, the worst result is annoyance of the noise. Explosions in the silencer or exhaust pipe can be obviated by reducing the oil supply, and are often caused by starting the engine before the igniting apparatus is sufficiently heated to cause proper ignition.

Leakage of Water.—Engines will sometimes refuse to operate due to this cause. Leakage of water can easily be ascertained by examination of the piston and cylinder, or the piston can be withdrawn from the cylinder. Testing of the water-jackets has already been explained in Chapter III., and the leakage, if found, must be remedied by new joints. If such leak-

age is due to defect in the casting, it can sometimes be remedied by drilling out the defective material and by tapping and plugging the cylinder walls or other defective part. This work, however, requires considerable care to thoroughly overcome the leakage.

CHAPTER X.

VARIOUS ENGINES DESCRIBED.

THE CROSSLEY OIL ENGINES

FIGURE 71 illustrates the Crossley oil engine having one heavy fly-wheel. Their "lampless" type of engine is shown in Fig. 71*a*, which has their latest vaporizer shown in section at Fig. 3 and two heavy fly-wheels suitable for electric lighting purposes. The method of vaporizing and igniting used with the Crossley engine is fully described in Chapter I. devoted to that subject.

The fuel oil-tank is placed against the cast-iron base of the engine, and the oil is pumped to the vaporizer in the usual way by an oil-pump actuated by the cam-shaft and in regular fixed quantities, but the fuel is allowed to enter the vaporizer only in exactly the proper quantity, the oil supply being controlled by the special measuring device, which consists of an inlet automatic valve leading to the vaporizer and an overflow-pipe leading back to the oil-tank. If the oil supply from the pump at any time is greater than the amount of oil which should enter the vaporizer, the fuel is re-

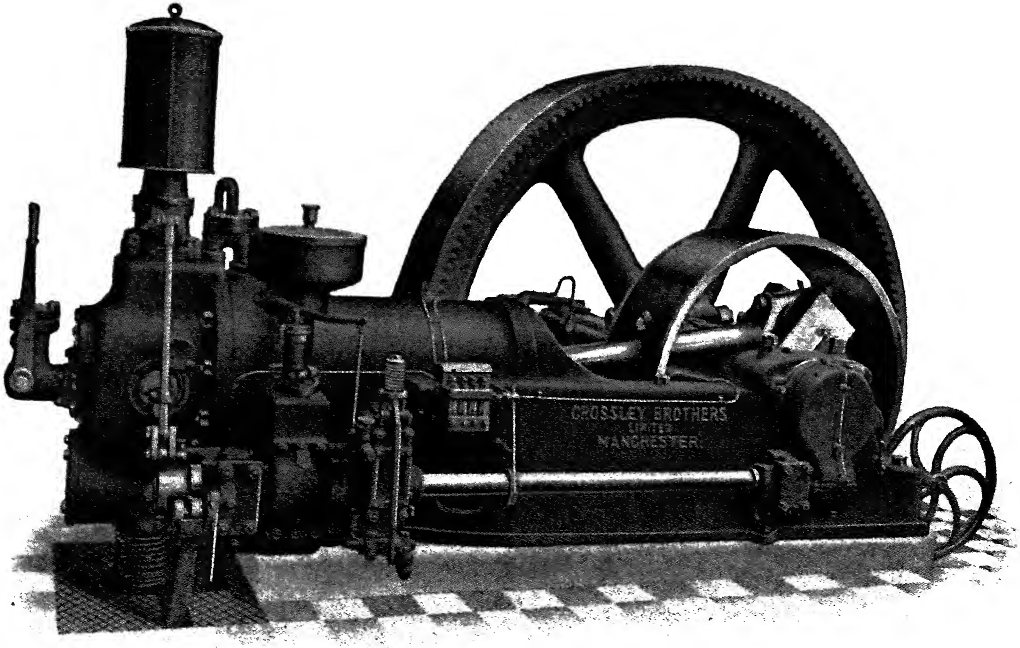
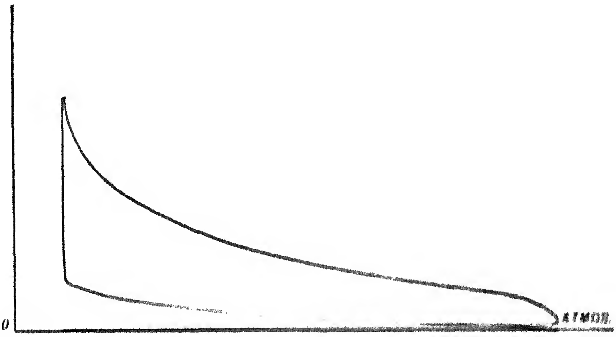


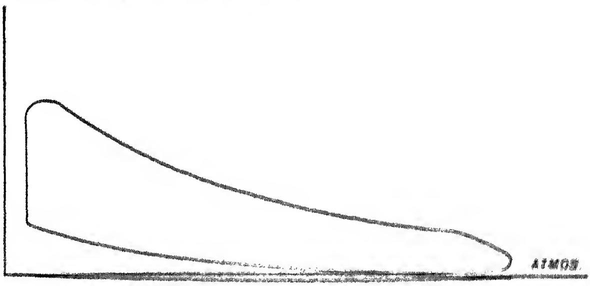
Fig. 71.

jected by the oil-measuring device, which is actuated by the partial vacuum in the cylinder during the air-



10

Diagram from the Crossley Engine: Revolutions per minute, 180; M. E. P., 60 lbs.; compression pressure, 48 lbs.; maximum pressure, 240 lbs.



11

Diagram from Crossley Engine: Revolutions per minute, 180; M. E. P., 50 lbs.; compression pressure, 50 lbs.; maximum pressure, 180 lbs.

suction period. The oil then returns through the overflow-pipe to the tank.

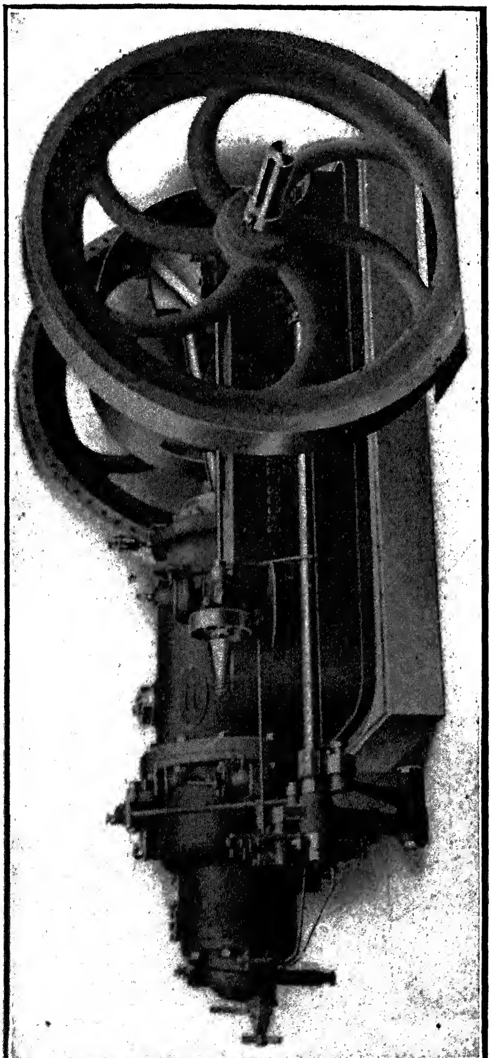


Fig. 71a.

The centrifugal governor is actuated by separate gearing and horizontal shaft direct from the crank-

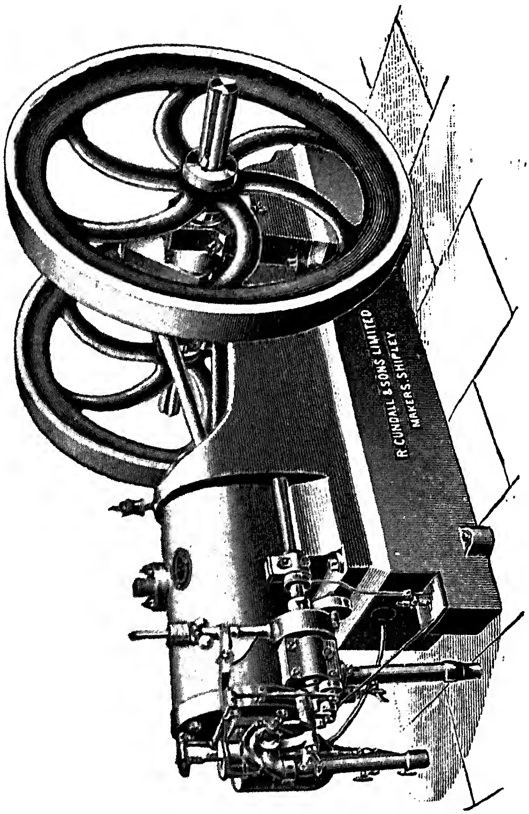


FIG. 72.

shaft, and the governor regulates the speed of the engine by acting on the hit-and-miss system, and con-

trols the vapor inlet-valve to the cylinder. Thus, if the required speed of the engine is exceeded, the vapor-valve is not opened, and accordingly only air is drawn into the cylinder through the air-inlet valve on the top of the cylinder, which is actuated by eccentric from the cam-shaft. No oil vapor is drawn into the cylinder, and the next explosion is missed. The lamp for heating the vaporizer receives its supply from the oil-tank placed against the base of the engine. The oil for the lamp is supplied by a separate pump, both oil-pumps being actuated from the same eccentric.

THE CUNDALL OIL ENGINE.

This oil engine is illustrated in Fig. 72, and it has oil-tank in the cast-iron base of engine, the fuel being pumped to the vaporizer in the usual way, the oil supply being regulated by a small adjustable thimble inside the cup on the vaporizer. The vaporizer and tube are heated by separate lamp supplied from oil-tank placed above the engine by gravity feed. Both air and exhaust valves are actuated from the horizontal cam-shaft in the usual way. The centrifugal governor is operated by bevel-gearing from the cam-shaft and controls the speed by acting on the oil-inlet valve.

THE CAMPBELL OIL ENGINE.

Fig. 73 illustrates larger-sized engine fitted with one fly-wheel only and outside bearing suitable for electric-

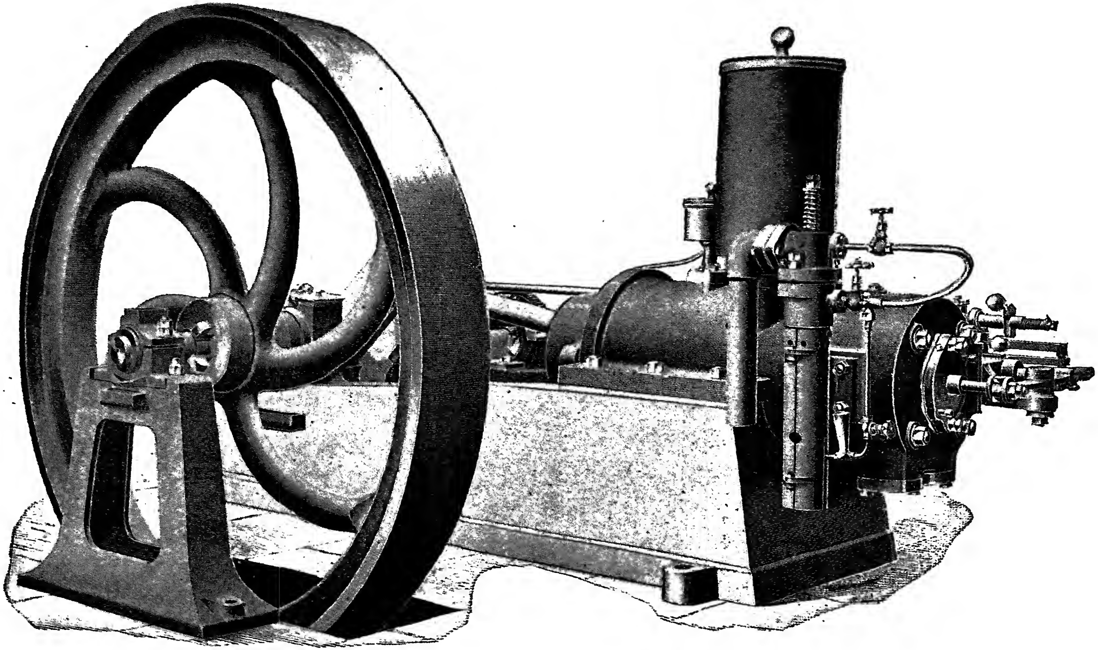
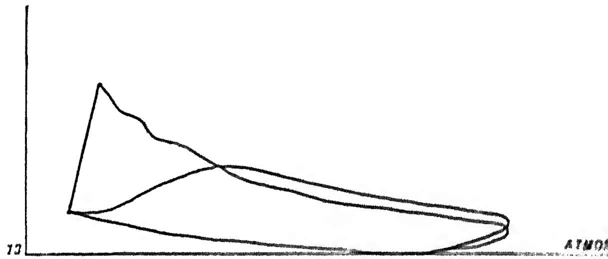
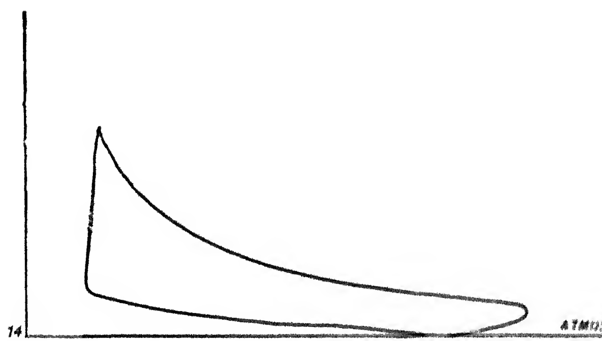


FIG. 73.

lighting purposes. The vaporizing and igniting apparatus of this type is described in Chapter I. The fuel



Light-load diagram taken from Campbell engine: Cylinder, 9.5″ in diameter; stroke, 18″; revolutions per minute, 210; M. E. P., 55.9 lbs.



Full-load diagram from Campbell Engine: Cylinder, 9.5″ in diameter; 18″ stroke; revolutions per minute, 210; M. E. P., 69.25; compression pressure, 55 lbs.; maximum pressure, 232 lbs.

oil-tank is placed on the top of the cylinder and the

fuel is fed by gravitation to the vaporizer and to the heating lamp, there being no oil-pumps. There are only two valves—the air-inlet valve, which is automatic, and the exhaust-valve, which is operated by the cam, which is actuated by spur-gearing from the crank-shaft, the necessary power to open the valve being transmitted through the horizontal rod in compression. The centrifugal governor is mounted on separate horizontal shaft, and is actuated by separate gearing from the crank-shaft. The speed of the engine is controlled by suitable device which is inserted by the action of the governor between the exhaust-lever and the stationary bracket when the normal speed is exceeded, thus holding open the exhaust-valve and preventing any of the oil vapor and air from entering the cylinder during the suction period.

Priestman Oil Engine.

Fig. 74 represents this type of engine as made by Messrs. Priestman in the United States.

The design of this engine is upon the " straight line" principle, and differs from the other engines herein described. In this engine, both the fly-wheels are arranged to be inside of the main shaft bearings instead of at each side of the frame, as is usual. The makers claim great advantages for this design, inasmuch as the strain on the bearings is minimized. The crank-pin is placed between the two fly-wheels, the hub of each fly-

wheel becoming the cheek of the crank. The oil is placed in the bed of the engine; an air pressu five or six pounds is always maintained in this tan means of the separate air-pump actuated from cam-shaft by eccentric. The vaporizer spraying igniting devices are fully described in Chapter I.

The governor is driven by belt from the crank-

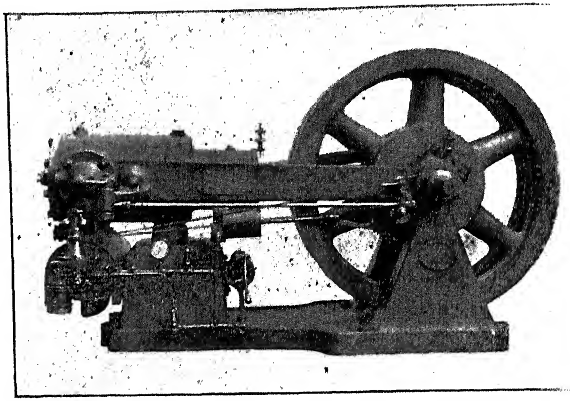
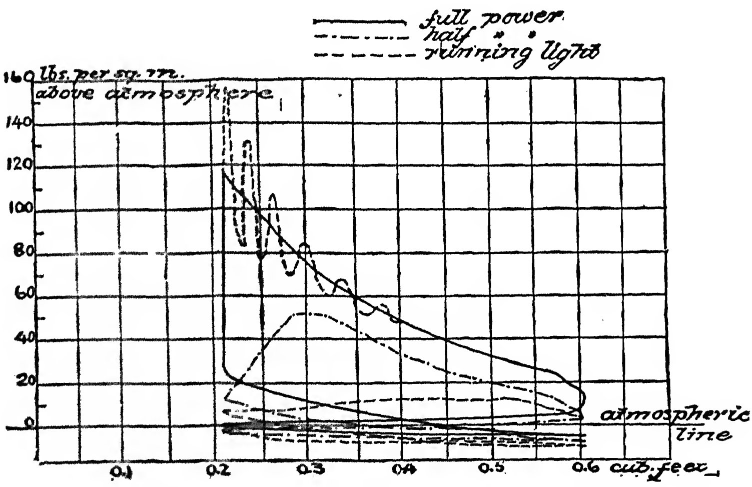


FIG. 74.

and is of the centrifugal or pendulum type. speed of the engine is controlled by suitable mecha acting on the throttle-valve regulating the supp oil and air entering the vaporizer. The air-inlet to the cylinder is automatic, the exhaust-valve actuated by horizontal rod operated from a cam p

on the cam-shaft. This engine, it is claimed, requires little or no lubrication for the piston.

The following test was made in the Engineering Laboratory at University College, Nottingham, England, on single-acting horizontal English type of Priestman oil engine having cylinder $10\frac{3}{4}''$ dia. and



INDICATOR CARD OF THE PRIESTMAN ENGINE.

$14''$ stroke, capable of developing $10\frac{3}{4}$ actual or brake horsepower at 160 R. P. M. The test was made after seven years' service of the engine using American kerosene, known as Royal Daylight, specific gravity 0.792 at 60° Fahr. and having flash point 83° Fahr. The effective work recorded is the effective indicated

pressure in the cylinder, the back pressure of the exhaust and suction strokes being deducted.*

TABLE V.

TRIALS OF PRIESTMAN OIL ENGINE, DEC. 9, 1900 (ROBINSON).

Duration of run (hours)	—	2
Revolutions per min. mean	—	160
Pressure before ignition (above atmosphere), lb. per sq. in.	—	20
Mean pressure, lb. per sq. in.	—	44
Mean back pressure (pumping strokes) lb. per sq. inch	—	3
Net effective pressure	—	41
Net effective indicated H.P.	—	10.5
Brake or actual H. P.	—	8.4
Engine friction H. P.	—	2.1
Mechanical efficiency per cent	—	80
Oil used per hour (total lb.)	—	8.82
" " " " per I.H.P. lb.	—	0.84
" " " " per B.H.P. lb.	—	1.05
Cooling water through jacket, lb. per min.	—	22.5
Cooling water rise in temp. 57° to 113° Fahr.	—	56°

THE MIETZ & WEISS ENGINE.

This engine is illustrated in Fig. 75. It works not, as some other engines described herein, on the Beau de Rochas cycle, but on the two-cycle principle—that is, an explosion is obtained in the cylinder at each revolution of the crank-shaft. As the oil-tank is above the cylinder, fuel is supplied to the smaller engines partly by gravitation—the quantity in-

*"Gas and Petroleum Engines," by Prof. Wm. Robinson, pp. 688.

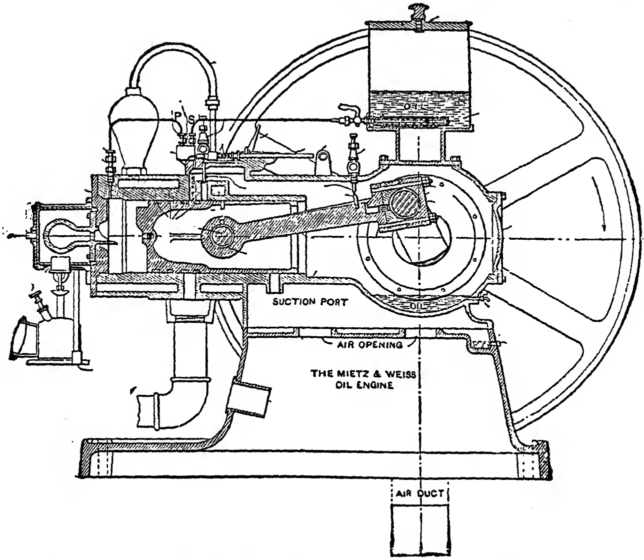


FIG. 75a.

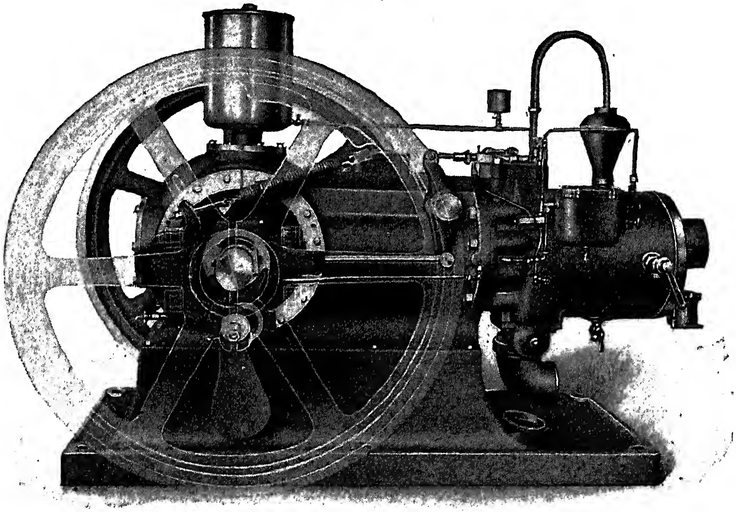


FIG. 75.

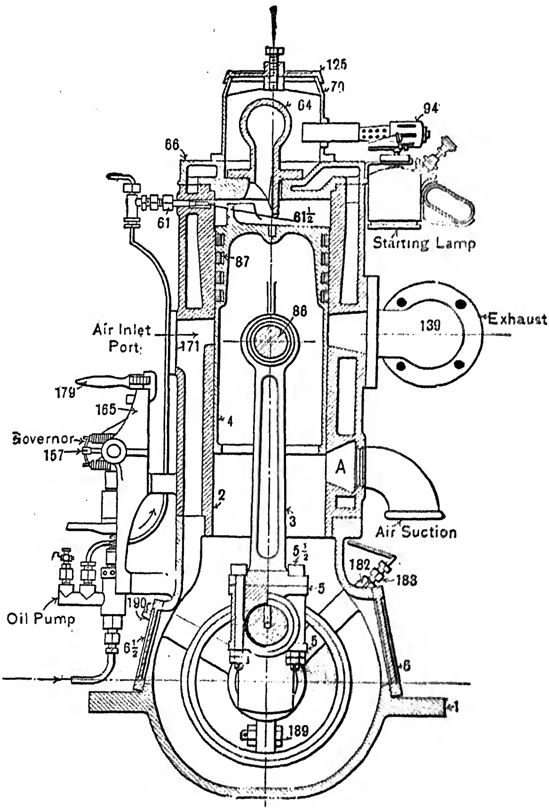

125
70
64
94
66
61
61½
Starting Lamp
87
86
Exhaust
139
Air Inlet Port
171
179
165
4
Governor
167
A
2
3
Air Suction
5½
182
188
5
190
Oil Pump
6½
5
6
1
189

FIG. 75C

jected, however, into the cylinder being regulated by small oil supply pump. Where required, the oil tank can be placed below the level of the engine. A sectional view of the horizontal engine is shown at Fig. 75*a*. The Mietz & Weiss marine engine is also shown at Fig. 75*c*, made vertical of single or multi-cylinder type. It operates on a similar plan of operation to the horizontal engine, a special feature of the multi-cylinder type being the use of one oil pump for the injection of the fuel into one or more cylinders.

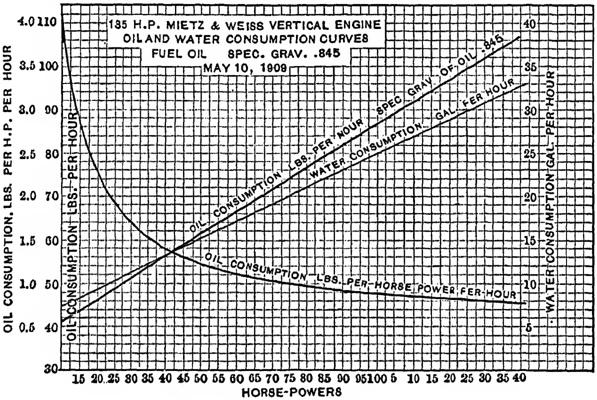


FIG. 71*b*.

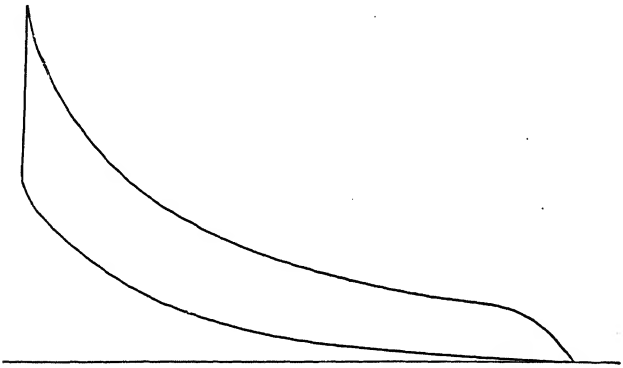
This vertical marine type engine is made in sizes up to 200 H. P., and is also used for industrial purposes direct-connected to electric generators and for general power purposes. The fuel is injected into the cylinder of the Mietz & Weiss engines with some steam. The steam being generated in the water jackets surrounding the cylinder, which are allowed to rise to a temperature necessary for generating the steam. The oil is vaporized in a hot chamber shown in the accompanying sectional illustrations placed at the back of the cylinder, which is heated for a few minutes in starting by independent lamp. Afterwards the heat

created by constant combustion maintain the igniter at proper temperature automatically.

The governor of the improved Mietz & Weiss engine is of the centrifugal type, and acts through a variable stroke on the kerosene pump, graduating the charge for varying loads. The governor weight is arranged near the shaft at the hub of the fly-wheel, to which it is pivoted at one end, the other end being secured to an adjustable spring, the tension of which determines the speed. The eccentric is free to slide at right angles to the shaft, and, being pivoted to the extreme end of the governor weight, receives a slight turning movement ahead from no load to full load. The regulation with this governor is extremely close in direct electric lighting service, where many of these engines are in use, either belted or direct-coupled to generators.

The deficiency of pressure in the crank-chamber is used to raise the lubricating oil from an oil well placed below the sight feed oilers which supply oil to the cylinder and crank-chamber. The crank bearings are lubricated by means of ring oilers. These engines are now made in various sizes from 1—200 HP, being direct-connected to dynamos, as shown in Fig. 58*a*. They are also direct-connected to centrifugal pumps, hoists as well as air-compressors. The compression of the air is generated in the crank-chamber and the air is drawn into the cylinder at a slight pressure during each outstroke of the piston. The exhaust opening is automatically uncovered by the piston, the exhaust passage being made in the cylinder wall. As the

piston travels toward the end of the stroke, this passage is uncovered, and the products of combustion are free to pass to the exhaust-pipe, while the



Indicator diagram taken from the Mietz & Weiss Engine: diameter of cylinder, 12″; stroke, 12″; revolutions per minute, 300; scale, 100; B. H. P., 20.

piston travels to the end of the stroke and the first part of the return stroke until the port is again covered, when the compression period commences for the next explosion. Consequently no valves are necessary, the air inlet to the cylinder being controlled by the action of the piston only, which simplifies the action of the engine.

Hornsby-Akroyd Oil Engine.

Fig. 76 shows this engine as made by the De La Vergne Machine Company, of New York. It is also made by the patentees at Grantham, England, and in France and Germany.

The Hornsby-Ackroyd engine is made in sizes of $1\frac{1}{2}$ to 500 H. P., all sizes being made of the horizontal type. This engine as made by the English makers is shown at Fig. 77. The fuel oil-tank is placed in the base of the engine and the fuel is delivered to the vaporizer by the small pump actuated from the cam-shaft by the lever which also actuates the air-inlet valve. The oil supply is raised to the vaporizer valve-box in regular quantities, but the oil is only allowed to enter the vaporizer to the required amount, the remainder of the oil flowing back to the tank through the by-pass valve which is regulated by the governor. Thus, if the speed of the fly-wheel exceeds the normal number of revolutions for which the engine is set, the governor mechanism opens the by-pass oil-valve, allowing part of the oil to flow back to the oil-tank, and accordingly reduces the charge entering the vaporizer, and consequently the mean pressure for one or more explosions is reduced in the cylinder. The governor is of the Porter type, actuated by gearing from the cam-shaft. The method of vaporizing and igniting is fully described in Chapter I. Both air-inlet and exhaust

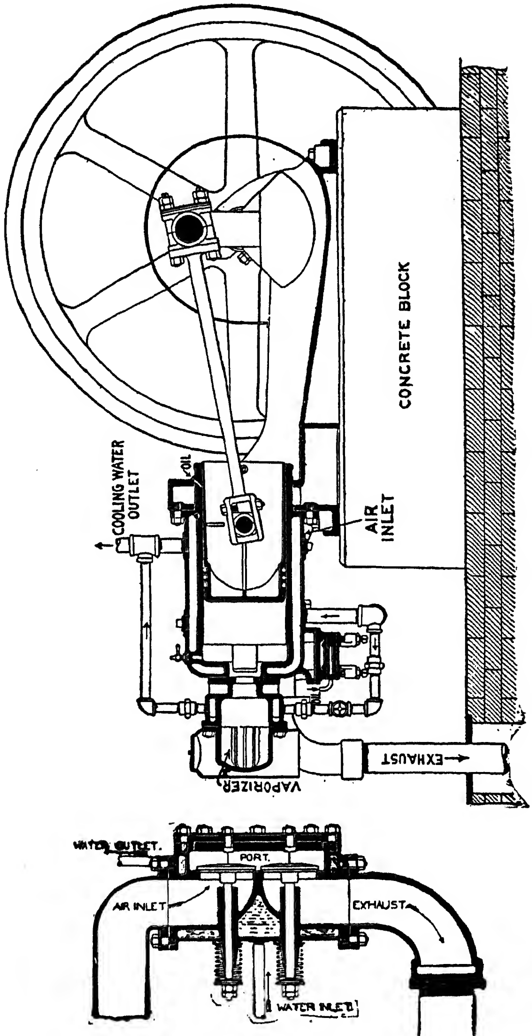

COOLING WATER OUTLET
OIL
AIR INLET
CONCRETE BLOCK
EXHAUST
VAPORIZER
WATER OUTLET.
PORT.
AIR INLET
EXHAUST
WATER INLET

FIG. 76.

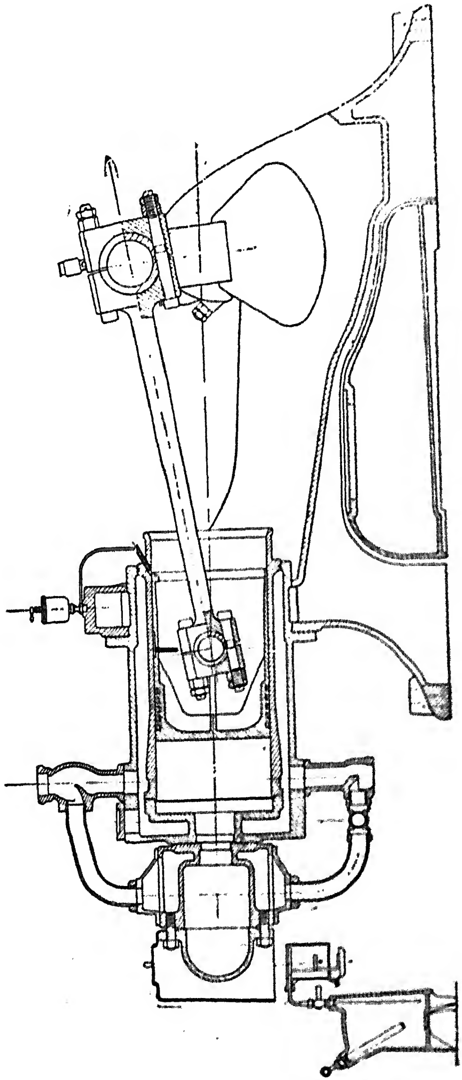


FIG. 77.

valves are actuated from the cam-shaft, these valves

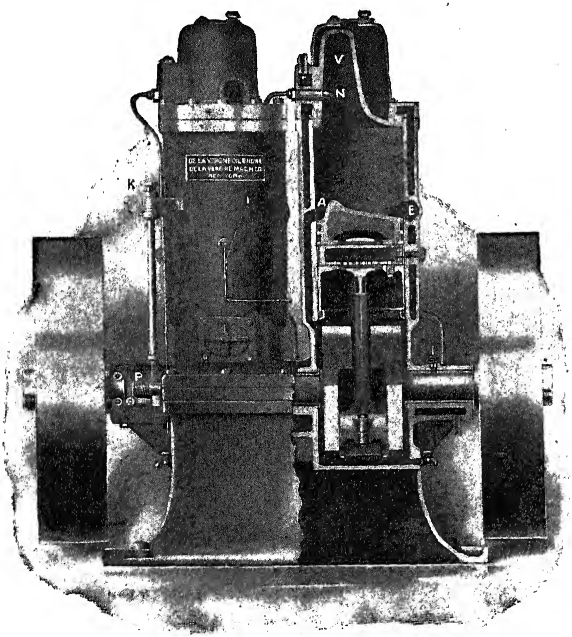


FIG. 77*a*.

being placed on the side of the engine. The air inlet in this type is different from the other engines de-

scribed. In this case the air enters not through the vaporizer, but by means of separate valve-chamber.

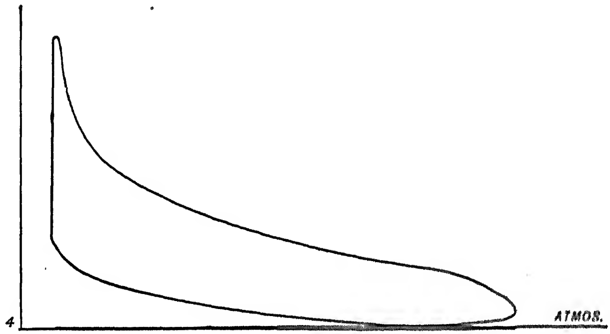


Diagram taken from Hornsby-Akroyd Engine: M. E. P., 48 lbs.; compression pressure, 50 lbs.; maximum pressure, 160 lbs.; revolutions per minute, 185; cylinder, 18.5″ diameter; 24″ stroke; full load.

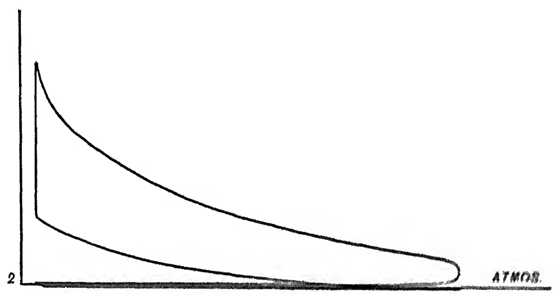


Diagram taken from Hornsby-Akroyd Engine: Diameter of cylinder, 11″; stroke, 15″; M. E. P., 49.5 lbs.; compression pressure, 60 lbs.; revolutions per minute, 230; consumption of oil W. W., 150° F. 0.8 lbs. per B. H. P. per hour.

A two-cycle vertical high speed engine is shown at Fig. *77a,* made and patented by the De La Vergne Machine Company. This engine operates on the two-cycle plan, as explained on page 17.

The features peculiar to this engine are the vaporizer, which is illustrated Fig. *77a* at V, and the sprayer, which is shown at N. This sprayer is also shown at Fig. *7a,* and described on page 13. As will be seen from Fig. *77a,* the vaporizer is made of a conical shape and the oil is injected directly into it.

The compression of the air before explosion takes place in the crank-case and enters the cylinder at passage A. There being no contracted opening to the vaporizer, and as a compression pressure of 100 lbs. is used, the clearance in the combustion space is very small and all the air entering the cylinder is forced into the vaporizer, where it freely mingles with the fuel. A baffle plate placed on the piston deflects the air into the vaporizer and a slight scavenging effect is produced, which forces the exhaust gases from the combustion chamber. The exhaust opening is shown at E.

The engine runs at approximately 500 R. P. M. and is specially adapted for direct connection to electric generators.

The governor is shown in detail at Fig. *24b,* and is of the centrifugal type placed in the fly-wheel, and is arranged to operate directly on the oil supply pump. The indicator cards are shown at Fig. *77b,* that at A being from the power cylinder at fuel load, and that at B taken from the crank chamber.

This engine is made in sizes up to 25 H. P. of the twin cylinder type. The bearings of the larger sizes are water-jacketed to insure maintenance of low temperature and allow free lubrication. Oiling of all bearings is effected by means of a force feed oil pump.

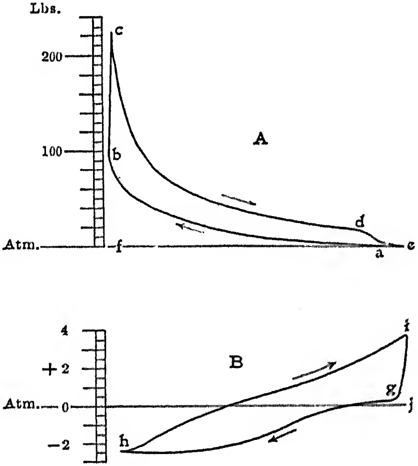


Fig. 77b.

The vertical type Hornsby-Akroyd engine, which was previously built, is also shown here in section (Figs. 78 and 79). The cam-shaft is operated

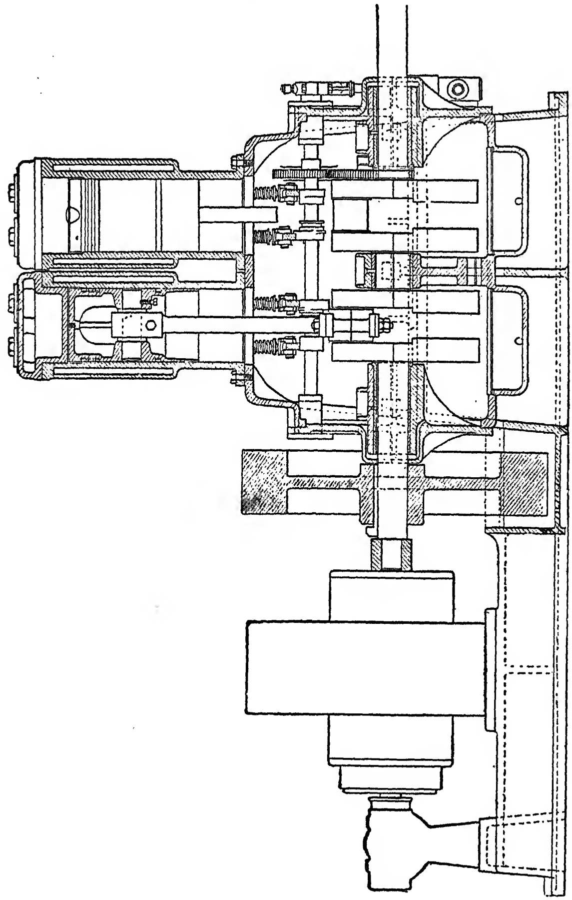


FIG. 78.

by a gearing from the crank-shaft in the regular way, the valves being operated by levers and rods. As will be seen from the illustration, the cylinders are built separately, being water-jacketed and mounted on a

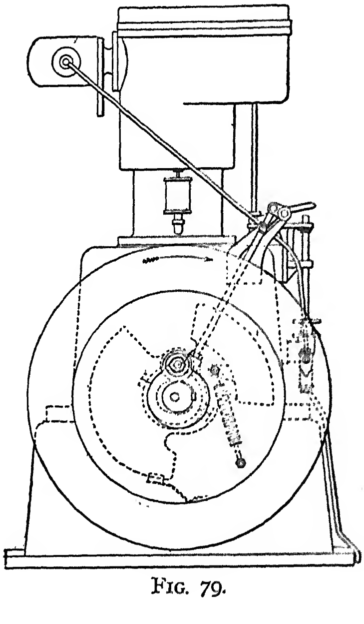


FIG. 79.

cast-iron frame of the enclosed type containing the crank-shaft. Lubrication is effected from the splashing of the crank in a bath of oil. The 15 H. P. engine has cylinders $8\frac{1}{2}''$ diameter by $9''$ stroke. The governing is effected by regulating the length of the stroke of the oil pump; no adjustment of the pump is therefore necessary. The governor is of the Rites pat-

ent type, and a regulation of less than 2 per cent is claimed by the makers of this engine, with a variation of the load within the engine's limits.

THE RITES GOVERNOR.

An illustration of the Rites governor is shown at Fig. 80. It will be seen that it is placed in the fly-

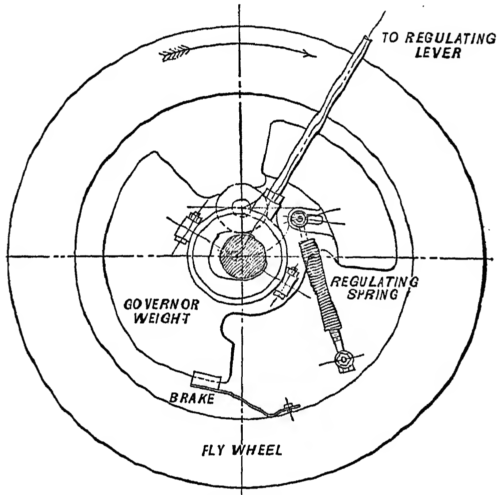


FIG. 80.

wheel in the usual way with this type of governor. The Rites governor has now become so widely known that only a short description is necessary. Briefly, it consists of but a single weight, distributed on opposite sides of the shaft with a spring connection to balance centrifugal force. In its application to the oil or gas

engine an eccentric cast in one piece with the weight structure is provided. The movement (while in operation) of the governor weight consequent upon any change in speed of the crank-shaft is transmitted to the regulating device by means of the eccentric attached to the governor weight, on which are fitted eccentric straps and rod. The other end of this eccentric rod is attached to a lever, which reciprocates the shaft on which is placed the eccentric fulcrum controlling the stroke of the plunger of the oil-supply pump or the opening of the gas valve.

The operation is as follows: If the speed of the crank-shaft is increased by a fraction beyond the required maximum speed, the momentum of the weight overcomes the strength of the spring, thus changing the throw of the eccentrics, which in turn reduces the length of the oil-pump stroke.

Among the many claims for the Rites governor are the following: It allows of a large range of adjustment. It is remarkably quick in action, and the distribution of the governor weights on each side of the weight-pin and also on each side of the crank-shaft allows the governor strength to be greatly increased without necessarily increasing the centrifugal element correspondingly, and utilizes the inertia action of the governor most effectively for extreme changes of load. There is only one bearing requiring lubrication—namely, that of the fulcrum pin. No dash-pot is required, and only a small brake or drag is used to steady the movement of the governor weight.

The speed of the engine is altered by the adjustment

of the spiral spring controlling the weights. Speed is increased by moving the pin holding spring outwards from the fulcrum pin and at the same time correspondingly increasing the tension of the spring, to preserve a constant proportional initial tension corresponding to the change of leverage of the spring.

To decrease speed, reverse the above operation, or, if desired, add to the weight, thus increasing its centrifugal force.

To remedy racing, move the spring connection to the governor weight in its slot *away* from the weight-pin, leaving the tension of the spring unchanged. If it is required to regulate closer, reverse this movement of the pin in its slot; that is, *towards* the weight-pin.

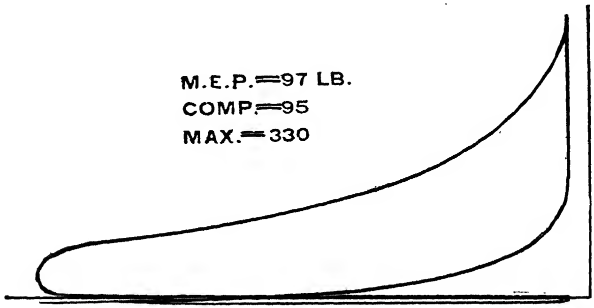


FIG. 80*b*.

JOHNSTON OIL ENGINE.—The Johnston oil engine is shown in Fig. 80*a*. It is made in various sizes up to 200 H. P. of the vertical type with one or more cylinders. It operates on the four-cycle principle, the air inlet and exhaust valves being actuated from a cam-shaft placed outside the crank casing operated by gearing from the crank-shaft in the usual way.

The chief feature of this engine is the method of ignition, which is effected by means of a hot surface, being a hot plate on the end of the piston, which is maintained at the proper temperature by the heat of combustion, and is insulated from the piston itself. (See Fig. 9.)

As will be seen from the indicator card at Fig. 80*b*, the compression pressure is approximately 100 lbs. per square inch, and the maximum pressure 300 lbs.

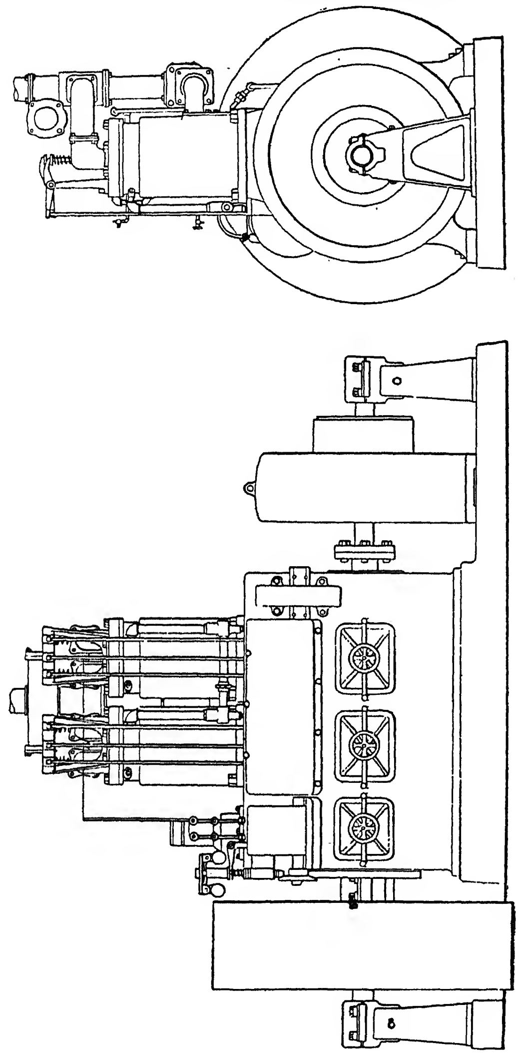
The injection of the fuel takes place after compression is completed, that is, at the end of the inward stroke.

A small air compressor attached to the crank-shaft furnishes the air necessary for spraying the fuel into the cylinder. The same compressor also furnishes the compressed air necessary for starting the engine. In starting, a metal thimble placed in the combustion chamber is heated by an external torch. An electric ignitor is used in some cases instead of the heated thimble for starting. The makers of this engine claim a fuel consumption of three-fifths of a pound of fuel or crude oil per actual B. H. P. per hour.

THE BRITANNIA CO.'S OIL ENGINE.

An engine fully described in the *Engineer** (London), made by the Britannia Co., of Colchester, England, is shown at Figs. 81, 82 and 83. It will be seen from the illustrations that it is of simple design. The vaporizer is a modification of the type as shown at Fig. 2 and referred to on page 8. The oil is stored in the base of the engine and is raised to the vaporizer by the suction of the piston. Consequently no oil pump is required. The air inlet valve *C* is automatic,

*See *Engineer and Engineering*, London, of June 19, 1903.



60 H.P ENGINE AS CONNECTED TO 35 K.W. GENERATOR

Fig. 80a.

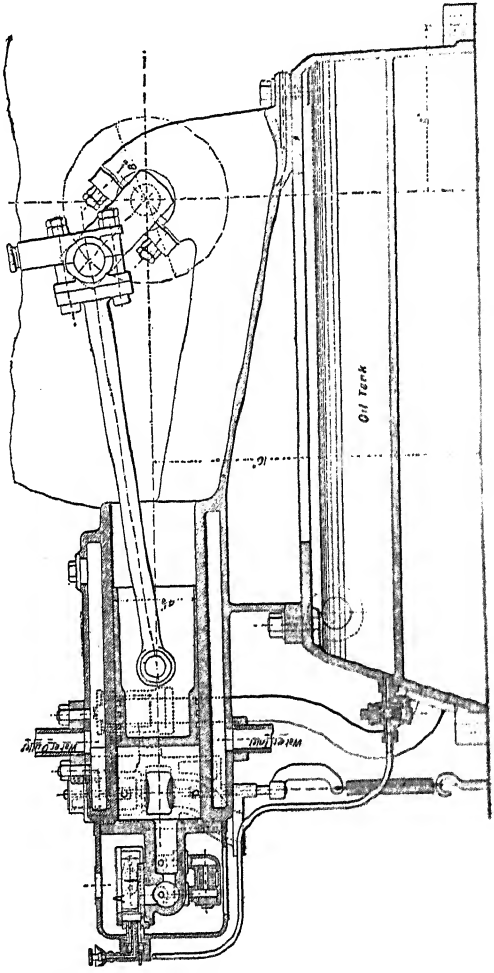


FIG. 81.

and is placed on the side of the engine above the exhaust valve *D*. The governor is of the centrifugal type and operates on the "hit-and-miss" principle, and is arranged to control the vapor inlet valve. On starting the engine the vaporizer is heated by external lamp for a few minutes and a small amount of fuel is injected into the vaporizer by means of the filling cup, marked *E*. The vaporizer consists of a flat cast-iron box, marked *A*, provided with baffle plates, which cause the oil or vapor to travel backwards and forwards

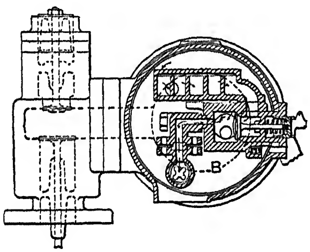


FIG. 82.

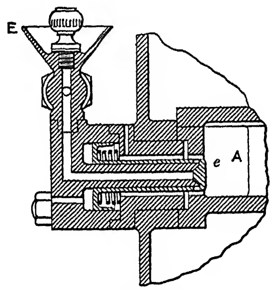


FIG. 83.

through passages before entering the cylinder. The ignition is caused by means of tube *B*.

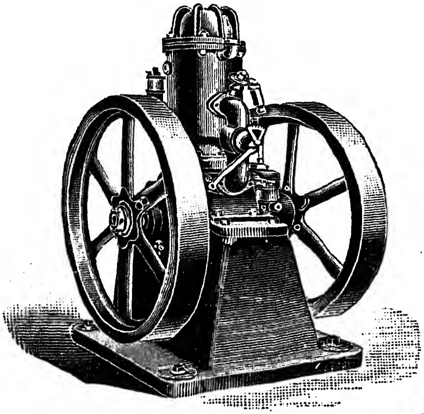
In operation the oil is raised to the vaporizer from the tank by the vacuum in the cylinder, caused by the outstroke of the piston. The cylinder communicates with the vaporizer through the vapor inlet valve only. Air enters both through the main air inlet valve *C*, Fig. 81, and a passage communicating with the vaporizer. The air entering can be throttled so that proportions of air entering by alternative ways can be regulated

as required. The oil supply enters by the passage closed by means of sleeve *c*, which forms also a valve as shown in Fig. 83. When the sleeve moves, due to the vacuum in the cylinder, by piston movement, oil is drawn (through holes in the sleeve) into the vaporizer. The amount of oil entering depends on the amount of air allowed to enter the cylinder through the vaporizer. When due to the action of the governor, the vapor valve remains closed, no communication is made with the cylinder and no oil enters the vaporizer. Two passages between the vaporizer valve and the cylinder are made, in one of which is the igniter-plug, which is simply a piece of steel having projecting internal ribs which absorb the heat from explosion, becoming red-hot in operation. No exhaust gases pass through the igniter, and on light loads gases only enter the igniter preceding an explosion. The temperature of igniter and vaporizer is easily maintained, and no stoppage due to the cooling of the vaporizer can occur.

American Oil Engine Co.'s Engine.

A vertical type oil engine made by the American Oil Engine Co., suitable for industrial and marine purposes, is shown in the single and twin-cylinder type at Fig. 84 and in section at Fig. 85. It is of the two-cycle type, the compression of the air previous to ignition being effected in the crank chamber, from whence it passes by a passage and port uncovered by the piston as it moves forward, to the cylinder. The fuel is supplied by oil pump operated by cam and

placed close to the sprayer shown in Fig. 85. The governing is effected by means of a sliding cam which



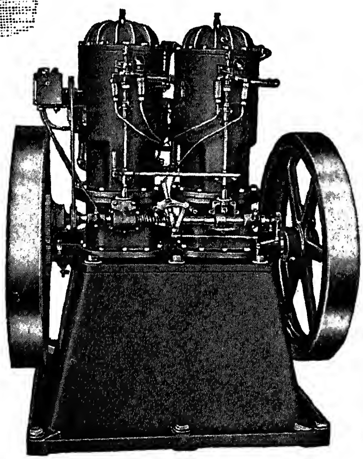


FIG. 84.

actuates the oil supply pump and shortens or lengthens the stroke of the pump in accordance with the load.

The ignition of the charge is caused by the heat of a steel disc on to which the fuel is sprayed. Starting is effected either with an electric igniter or by means of

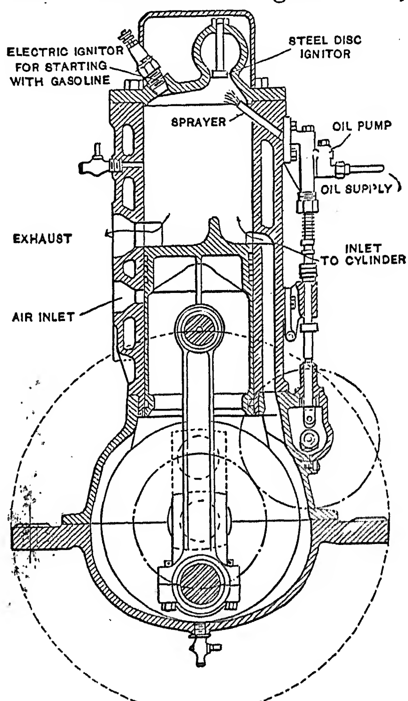


FIG. 85.

tube heated externally by kerosene torch. Gasoline or alcohol is used instead of kerosene for starting when the electric igniter is operated. A multiple force feed

oil pump furnishes lubrication to the cylinder and all bearings. This engine is made in various sizes from $1\frac{1}{2}$ H. P. upwards.

THE BARKER ENGINE.

A type of engine which in recent years has received some attention from inventors is that in which the cylinders revolve around a fixed crank-pin or cam. For situations where space is limited and where vibration should be eliminated and weight per horse power reduced to a minimum, the advantages of this type of engine are apparent.

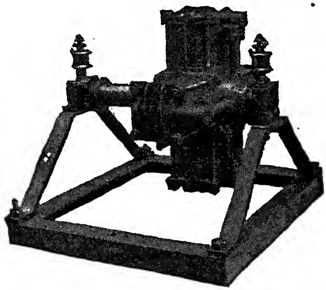


FIG. 86.

Fig. 86 shows the engine complete. It will be noted that there is no fly-wheel, the cylinders themselves

revolving around the centre bearing and furnishing the necessary momentum. The engine works on the "Otto," or four-cycle; that is, each cycle of operation in each cylinder consists of four strokes; thus a four-cylinder engine has four impulses each revolution. This is effected by the use of the cam motion shown in Fig.

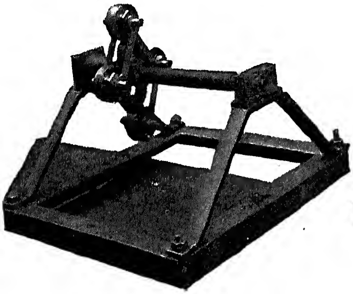


FIG. 87.

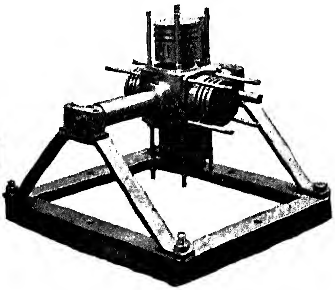


FIG. 88.

87, instead of the ordinary crank. This mechanism is equivalent to a double-throw crank.

Fig. 88 shows the four pistons in position, the cylinders having been removed.

The air and vapor inlet to the cylinders and the exhaust outlet are effected through the hollow spindle on which the cylinders revolve, radial ports or passage-ways being made in the spindle, which are uncovered by recesses in the cylinders, as these recesses coincide with the ports of the cylinder at each revolution.

The ignition is caused by electric igniter of the jump-spark type. The timing of the ignition is obtained by

a revolving contact breaker. When using gasoline, a carburetor of the ordinary float type is attached. When kerosene is used as fuel, a vaporizer somewhat similar to that shown at Fig. 3 is used, the heat from the exhaust gases being sufficient to maintain the required temperature for vaporization. The oil is fed by gravity and the vapor is drawn into the cylinder by the piston displacement in the usual way. The power is taken off from a pulley attached to the sides of the cylinder.

A motor of this type of one actual horse-power weighs about 15 lbs.; a 3 H. P. weighs approximately 35 pounds. A speed of about 800 R. P. M. is obtained, which speed is varied by the lead given to the igniter. When running at a high speed the engine can be held in the hands without vibration.

CHAPTER XI.

PORTABLE ENGINES.

PORTABLE type oil engines, made by nearly all makers of fixed horizontal engines, are used for various purposes. Such engines combined with air compressors are very useful for operating pneumatic tools used in structural iron work, repairs and similar work where compressed air is required in different locations for short periods of time. For portable electric-lighting purposes the oil engine (Fig. 89) is well adapted. Electric lighting outfits of this kind have been found useful for operating search-lights for military purposes and for supplying current for electric lighting for contractors, etc., where illumination of a portable nature is required for a short period only. The portable oil engine is also largely used for farm work, such as operating threshing machines, etc.

In all cases these engines are required to be frequently removed from place to place, and therefore must be as light as possible in design, but must be of such substantial construction that they can be transported from place to place over rough, uneven roads, and all provision for operation in the open air must be made. In Europe the portable engine is generally constructed somewhat differently to the ordinary fixed

engine. The heavy cast-iron bed-plate used in fixed engines is replaced with light steel construction, which considerably reduces the weight. This type of construction is shown in Fig. 89, and while it is somewhat more expensive than those portable engines composed of the fixed engine without base-plate bolted to steel or wooden truck, the advantage of lightness is gained as well as facility in transportation.

In the United States the portable engines are more generally composed of the standard fixed engine placed on steel or timber truck. This outfit is cheaper in cost than that of the special construction above mentioned.

The portable engine is often required to operate in localities where running water is not available, and therefore it must be self-contained as regards the cooling of the cylinder. An important feature of this outfit is, therefore, the cooling-water apparatus. In order that only a small amount of water may be used, different devices have been constructed for rapidly cooling a small amount of water. Such device in the Hornsby-Akroyd consists of a gradirwork placed inside the circular chamber, seen in Fig. 89, placed in the front of the engine. The water is circulated around the cylinder of the engine by a small reciprocating pump operated from the cam-shaft, and after passing through the cylinder and taking up the heat is delivered to the upper part of this chamber and flows down a wooden gradirwork. A draft of air is at the same time induced by the exhaust emitted above, which

rapidly cools the water as it trickles down the slats of the gradirwork. For a 20 H. P. engine only about 30 to 40 gallons of water are required.

Another device for cooling the water is that composed of trays over which the water flows while a

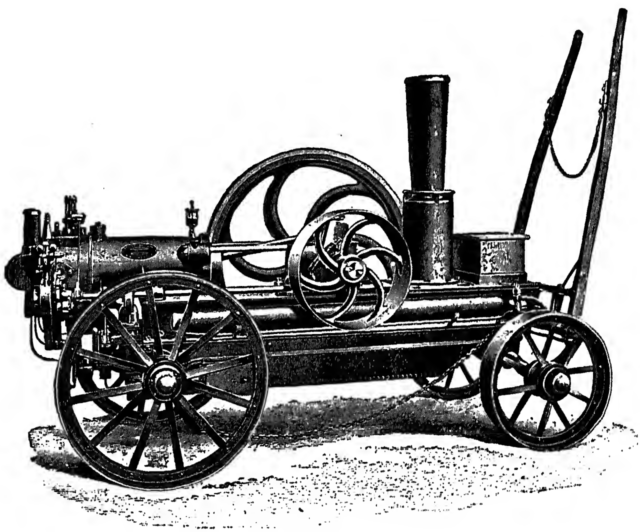


FIG. 90.

draft of air is induced in the same way as above mentioned.

An engine equipped with this cooling device is shown in Fig. 90, as made by Crossley Bros., Manchester, England.

Another type of portable engine is that shown in Fig. 91, consisting of the Mietz & Weiss engine. This

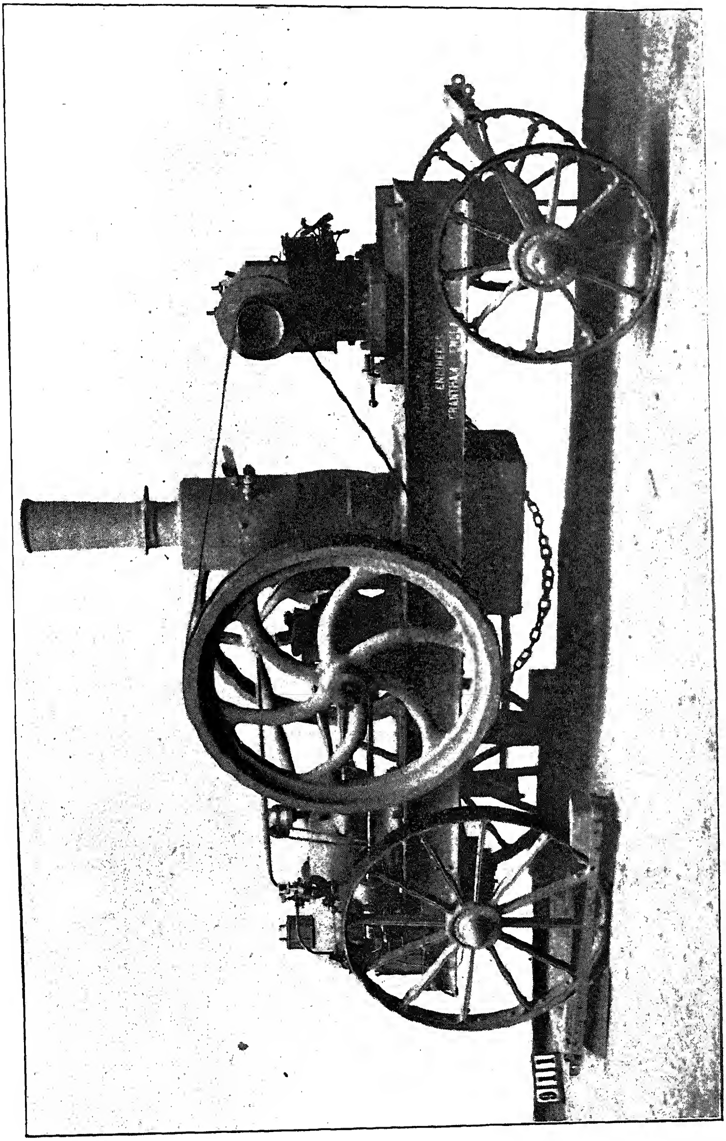


Fig. 89.

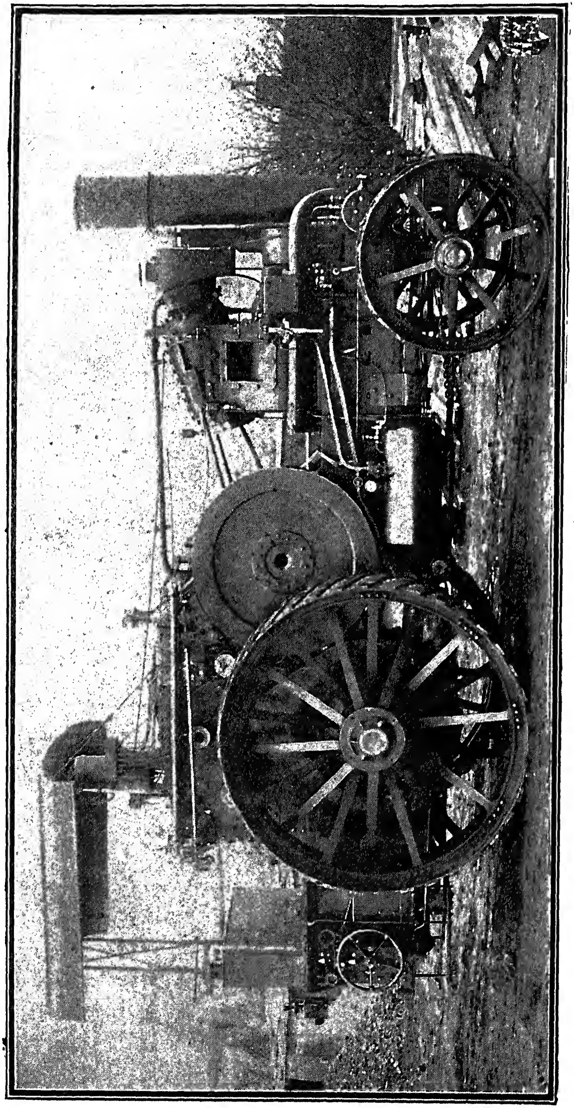


Fig. 92a.

is the standard fixed engine placed on a truck, the cooling water being supplied from a tank in front of the engine.

As the internal combustion engine cannot be balanced as effectually as the steam engine, greater vibration of the engine has to be overcome in holding it in

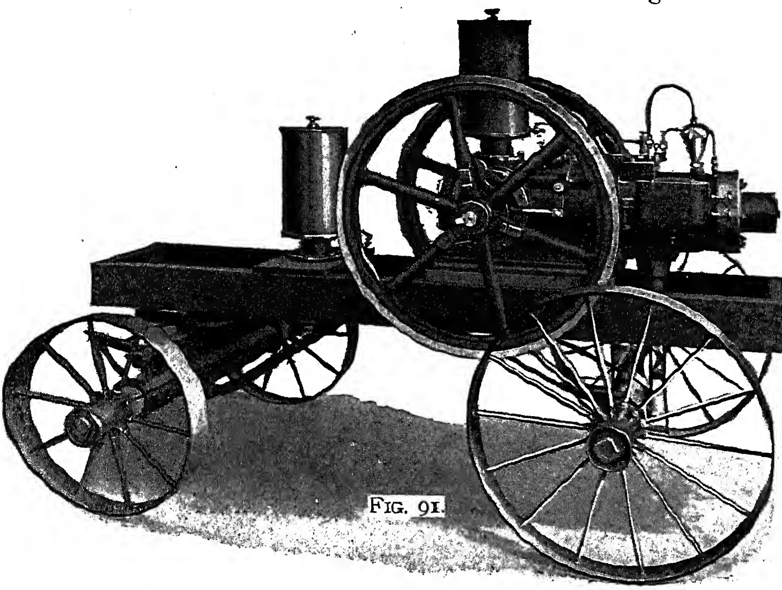


FIG. 91.

place. An important feature of the portable engine, therefore, is the chocking device which is required to hold it rigidly in position when in operation. In some engines simply a wooden chock is used, placed each side of the wheel and drawn together, holding the wheels from moving. A very effective device is that composed of four adjustable struts, each having turnbuckle fitting

into a flat timber plank placed on the ground lengthwise under the engine and protruding from each end. When it is desired to hold the engine in position,

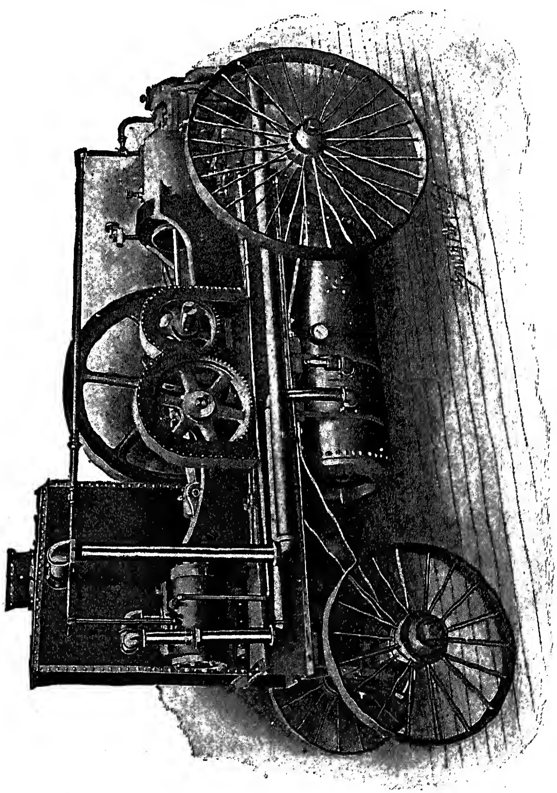


FIG. 92.

the struts, placed at each end of truck, are lengthened by means of the turnbuckle, thus taking the

eight off the wheels. By this means the engine is ɜld as rigidly as when on a concrete foundation, and ithout movement. When it is required to remove the ıgine the struts are shortened by simply unscrewing ıtil the weight is taken up by the wheels. The wear ı the wheels due to the continuous vibration of the ıgine is thus avoided, and the wheels can consequent- be lighter in construction.

A portable air-compressing outfit is shown in Fig. 2. As will be seen from the illustration, it ; composed of the oil engine, which operates the air- ɔmpressor by a gearing, the air receiver being placed eneath the frame of the truck, while the cooling-water evice is placed lengthwise with the air compressor.

An oil traction engine is shown at Figure 92*a*, in hich the ordinary frame and truck of the steam trac- on engine is used, the boiler being replaced by an il engine.

The engine shown in the illustration has two cylin- lers placed at an angle to each other, the connecting ods operating on one crank-pin, the power from the rank-shaft being transmitted by gearing to the road- vheels. The cooling of the water is effected somewhat similarly as with the portable engine. This type ɔf engine, made by Messrs. R. Hornsby & Sons, Grantham, England, after very severe tests recently received a first prize of £1,000 from the British War Department.

CHAPTER XII.

LARGE-SIZED ENGINES.

THE higher thermal efficiency of the gas engine as compared with that of the steam engine and its adaptability to use the poorer and cheaply produced gases made in the producer plant, the Mond gas plant, as well as the gases given off from blast furnaces, etc., has resulted in its development and manufacture in units as high as 5000 H. P.

The "oil gas" producer, an apparatus for furnishing gas produced from vegetable and mineral oils, is also used in connection with the gas engine; and also, as described hereafter, the apparatus developed by C. C. Moore & Co., of San Francisco, for generating gas from crude oil, which gases are furnished to the gas engine. Until recently the oil engine self-contained, that is, requiring no outside gas-making apparatus, of 100 H. P. was probably the largest unit made. The oil engine up to 500 H. P. is now, however, being manufactured.

The production of great quantities of petroleum in Texas and California chiefly useful for fuel purposes only, and which can be procured at a low price as compared with illuminating oils, has enabled the oil engine in many locations to compete in cost of installation and

operation with gas engines using producer and other cheap gas.

With the smaller size oil engines simplicity of construction is probably the most important feature, as it must be adapted for successful operation in the hands of unskilled attendants and be free from all delicate mechanisms which may require skilled attention. With the larger size engines, which have a greater earning capacity and which allow of the expense of a skilled attendant, simplicity of construction is not so important a feature. Mechanisms which may frequently give trouble in the smaller engines when in the hands of unskilled and inexperienced attendants may in the hands of the engineer attending to the larger engines give continuous satisfaction.

The tendency in design of the larger size gas engines is resorting to the two-cycle method of operation. Where the four-cycle method is adhered to two or more cylinders are employed. The four-cycle single-cylinder engine, as already explained in Chapter I., obtains an impulse once in two revolutions, and consequently during the three idle strokes of the piston the power and speed must be maintained by the momentum of the fly-wheels, necessarily enormous in an engine of 100 H. P. or over for the power obtained, in comparison with the fly-wheel of a steam engine of the same capacity. With the two-cycle engine, in which an impulse is obtained each revolution of the crank-shaft, double the power is developed as compared with the four-cycle engine of the same size. The mechanical efficiency is increased, owing to the reduced

weight of the fly-wheels, and the weight and cost of the engine per H. P. is curtailed.

The difficulty of procuring proper combustion in the two-cycle oil engine, more essential where crude oil is used than where gas or gasoline is the fuel, is not yet entirely overcome.

It has been previously stated that the larger size oil engines, to compete with the gas engine in cost of fuel, can do so only when a cheap grade of oil is used as fuel. To use such fuel, it is imperative that proper combustion takes place in the cylinder.

It is of interest to compare the relative cost of operation of the steam engine, the gas engine and the oil engine of, say, 50, 100 and 200 H. P. As the cost of fuel varies in different localities according to the cost of transportation, etc., this cannot be done to suit all cases. The following table, however, shows the relative cost of installing and operating a steam, gas and oil engine plant of 50 to 200 H. P. The cost of the plant includes cost of land, building of engine and boiler house, foundations, smoke-stack, etc., and all auxiliary apparatus. The cost of producer plant, and the cost of oil storage tanks and cost of apparatus for handling fuel is also included. It will be noted that the cost of water supply has in each instance been neglected. This is done because the amount of water required would be approximately the same with each type; possibly a saving in favor of the oil and gas engine would in many installations be effected. The figures must be modified to suit the actual cost of fuel in a locality differing from those given. The saving favorable to the gas-engine

TABLE VII.—RELATIVE COST OF INSTALLING AND OPERATING POWER PLANTS BURNING DIFFERENT FUELS.

Basis for Calculations: { 300—10 hour working days per year. Coal at $3.00 per 2000 lbs., or 0.15 cent per lb. Illuminating gas at 80 cents per 1000 cubic feet. Crude Oil at 3.5 cents per gallon of 7.5 lbs., or 0.4⅔ cent per lb.

Brake Horsepower.	50			100			200		
Type of Engine Used.	Steam Automatic Non-Condensing.	Illuminating-Gas Engine	Oil Engine.	Steam. Automatic Condensing.	Producer-Gas Engine.	Oil Engine.	Steam. Compound Condensing.	Producer-Gas Engine.	Oil Engine.
Cost of plant complete, with machinery, foundations, building and land, per H.P., $	160	130	135	145	150	120	120	132	102
Fixed charges, 16%* per H.P., $	25.60	20.80	21.60	23.20	24.00	19.20	19.20	21.12	16.32
Fuel per H. P. per hour......	7 lbs. coal	20 cu. ft.	0.9 lb.	6 lbs. coal	1.5 lb. coal	0.9 lb.	4 lbs. coal	1.25 lbs. coal	0.9 lb.
cents	1,05	1.60	0.42	0.90	0.225	0.42	0.60	0.1875	0.42
Fuel per H. P. per year.	21,000 lbs.	60,000 cu. ft.	2,700 lbs.	18,000 lbs.	4,500 lbs.	2,700 lbs.	12,000 lbs.	3,750 lbs.	2,700 lbs.
$	31.50	48.00	12.60	27.00	6.75	12.60	18.00	5.63	12.60
Oil, waste, supplies per H.P. per year ....$	2.70	3.50	3.50	2.40	3.30	3.30	2.00	3.00	3.00
Cost of attendance per H. P. per year.................$	18.00	5.00	5.00	10.00	7.50	5.00	10.50	8.00	5.00
Cost of one B.H.P. per year, $	77.80	77.30	42.70	62.60	41.55	40.10	49.70	37.75	36.92
Cost of one B. H. P. per hour................cents	2.593	2.577	1.423	2.087	1.385	1.337	1.643	1.258	1.231

The fuel consumption here allowed includes *all* the fuel used in a plant. It is based on the actual H. P. delivered, and represents average values as obtained under ordinary conditions of working.

* Fixed charges include: Interest on investment, 6%; depreciation and maintenance of machinery, 6%; building, 2%; insurance and taxes, 2%.

installation due to the recovery of by-products which is effected with the Mond gas plant is neglected, and should be taken account of where this system can be

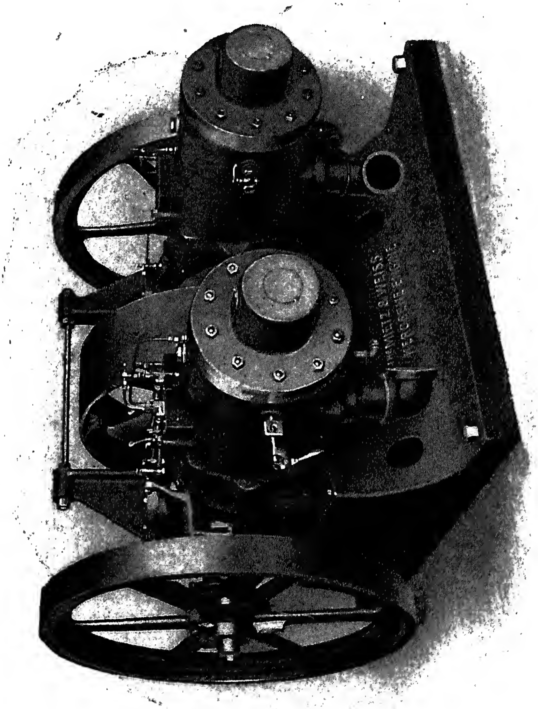


Fig. 93.

used. The steam turbine, it will be noted, is not mentioned in this classification, the steam engine considered being the reciprocating type.

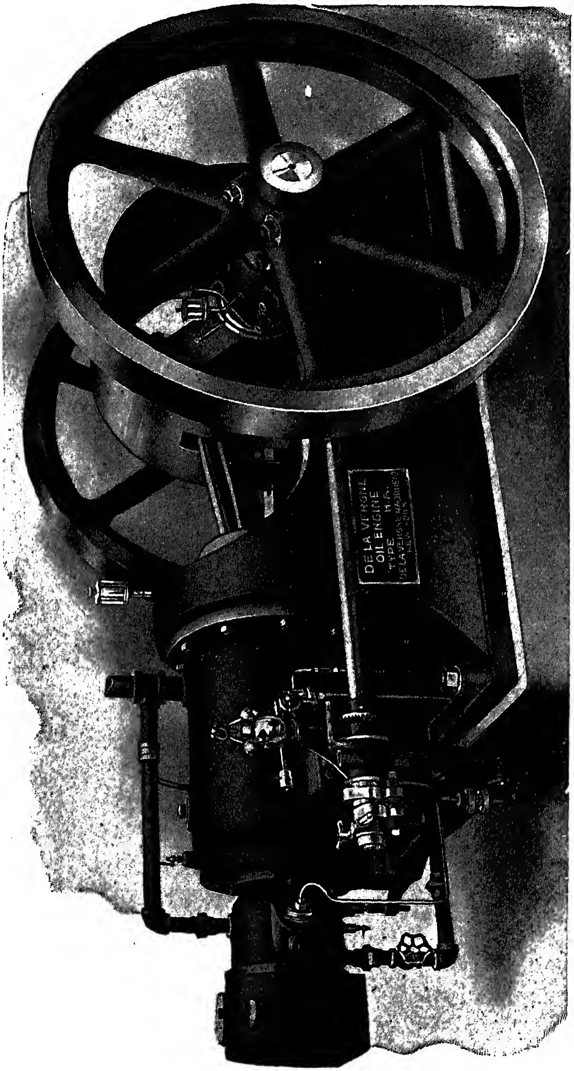


FIG. 94

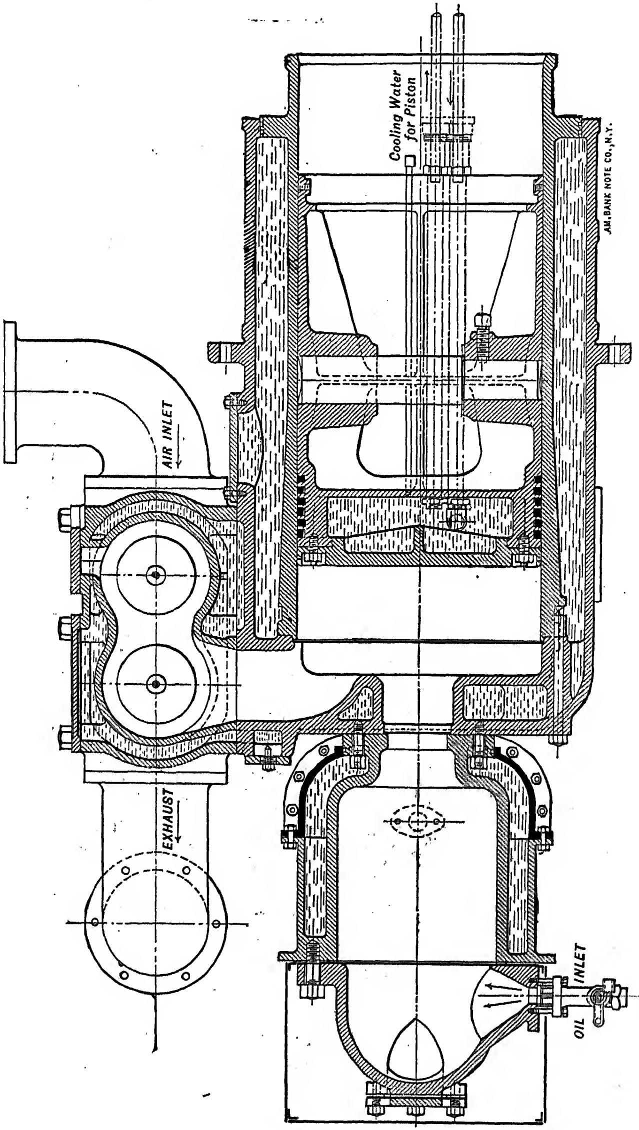

Cooling Water
for Piston
AIR INLET
EXHAUST
OIL
INLET
AM. BANK NOTE CO., N.Y.

FIG. 95.

THE MIETZ & WEISS two-cycle oil engine has already been described. An engine of this type of 60 H. P. is shown at Fig. 93. It will be seen that it consists of two smaller engines coupled together and placed on one base-plate. Each engine is self-contained and, if necessary, can be operated alone by simply uncoupling the connecting-rod, etc.

THE HORNSBY-AKROYD engine of 125 H. P. is shown in Fig. 94. This engine operates on the four-cycle system. Its proportions are necessarily large as compared with the two-cycle type, and, owing to the three idle strokes present when the Otto cycle is used, the fly-wheels must be very heavy to obtain even running. The advantage, however, is gained of obtaining a good combustion, which is not always the case with the two-cycle engine, and consequently crude oil can be satisfactorily consumed in this engine. The deposit of carbon when using crude oil is abstracted from the vaporizer through the hole in the back of that chamber shown in the illustration, and which is covered by a flange. These engines are now made up to 500 H. P. by R. Hornsby & Sons, Grantham, England.

A sectional view of the cylinder is shown at Fig. 95, in which will be noted the water-jacketed piston and the method of supplying the water to it. In other respects this engine operates in a similar method to the smaller sizes already described. They are started by compressed air supplied from a reservoir, the air entering the cylinder by means of valves and valve-box connected to the reservoir already described on page 105. In the larger engines water-jacketing of the pis-

ton is required in addition to the water-jacketing of the cylinder to preserve the proper temperature necessary for lubrication, and to prevent undue expansion of the piston being exposed to the greater volume of gases in the cylinder. The water is introduced by a sliding tube to the piston, with which it reciprocates.

The Diesel Engine.

The Diesel engines are built by the American Diesel Engine Co., at Providence, R. I. They are also built by several manufacturers in Europe, both in Great Britain and Germany. The Diesel engine, as at present made in the U. S. A., is shown at Fig. 96. The engine here described is the type built by the makers under American and Canadian patents.

The chief characteristic of the Diesel engine is the high thermal efficiency obtained and the consequent low consumption of fuel. The high thermal efficiency, which it is claimed is 38%, is due to the high compression of the air in the cylinder, to the exceedingly small clearance in the cylinder, which is approximately 7% only of the total cylinder volume, and to the slow combustion of the fuel which is effected by the method of injecting the fuel peculiar to the Diesel engine.

As will be seen from the accompanying illustrations, this engine is of the vertical type and is of very substantial construction. The cylinder walls, cylinder head and valve chambers are water-jacketed. The enclosed crank-chamber is advantageously made readily

accessible by means of removable plates on either side of it.

Fig. 97 shows in plan and partly in section the Diesel engine of the three-cylinder type. It is also made with single and double cylinder.

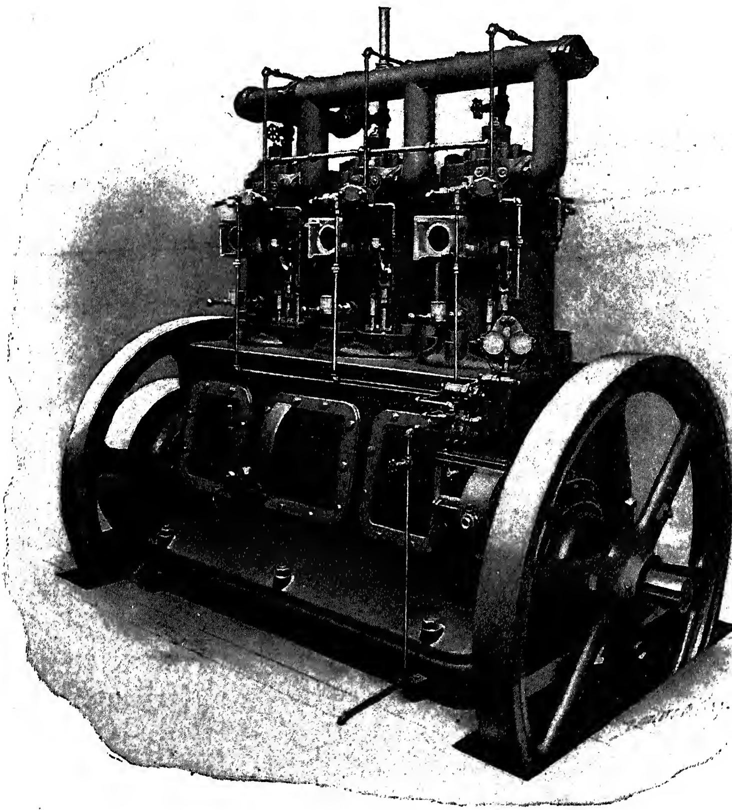


FIG. 96.

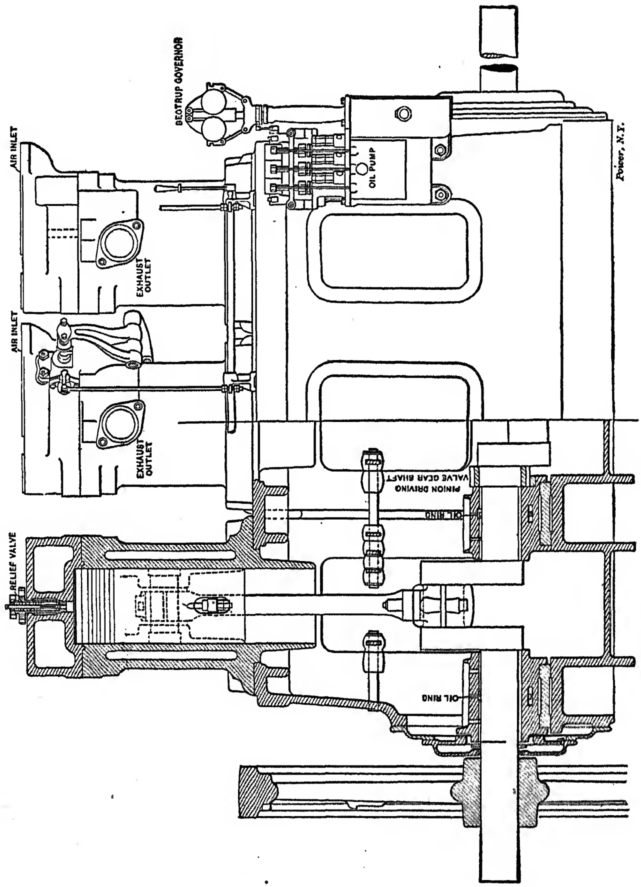


FIG. 97.

A sectional end view is shown at Fig. 98. The crank-shaft, or main bearings, are adjustable by means

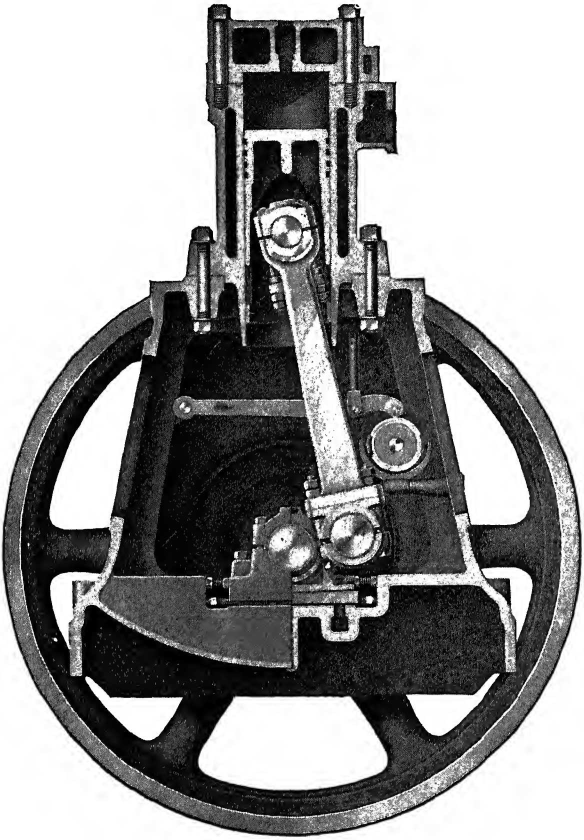


FIG. 98.

of wedges and screws, as shown. The piston is made as long as possible, in order to give a maximum bearing surface, and is fitted with steel snap-rings. The connecting-rods are of the marine type, with adjustable bearings at both ends. The valve motions are operated from the cam-shaft inside the enclosed frame, which is actuated by gearing from the crank-shaft. The engine operates on the "Otto," or four-cycle, principle. The air supply for supporting combustion is drawn into the cylinders through the air inlet valves placed in the housings to one side of the top of the cylinder head. (See Fig. 99.) The fuel to the cylinders is supplied by a separate oil pump for each cylinder. The oil pump is operated from a shaft geared to the cam-shaft. The method of operation is as follows:

The engine is first started by means of compressed air, which is supplied from an auxiliary air receiver suitably connected to the cylinder by means of a starting valve operated by a starting cam, thrown into action by hand, before starting. By this means compressed air is admitted to the cylinder and the piston is moved forward for one or two revolutions. Simultaneously compression of the air in the other cylinders takes place, which is sufficient to ignite the charge of oil in them. As soon as the ignitions take place the starting cam is automatically thrown out of action, the exhaust cam being simultaneously thrown into action. The admission valve for fuel and air under pressure is shown in Fig. 99. As will be seen, the valve spindle is surrounded by a series of brass washers perforated with small holes, being parallel to the

spindle. The fuel before entering the cylinder occupies the cavities in and between these washers as it is delivered from the fuel pump. Compressed air is introduced behind the oil inlet and at the opening of the

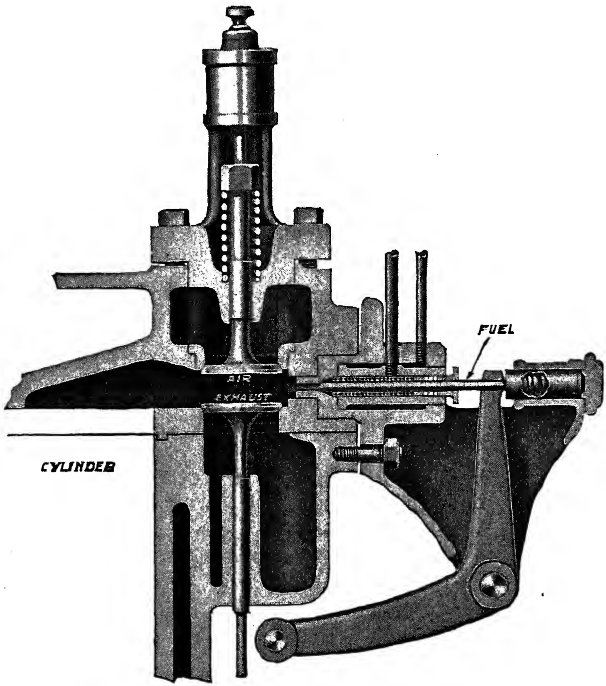


FIG. 99.

admission valve the oil is sprayed into the cylinder. The fuel enters the cylinder only after the compression stroke is completed and when the piston is beginning

to descend. The compression in the cylinder caused by the previous up-stroke of the piston reaches a pressure of 450 to 525 lbs. per square inch; resulting temperature approaches 1000° Fahr., which is more than sufficient to ignite the oil vapor. The fuel valve remains open about one-tenth of the period of the expansion stroke. The amount of fuel entering depends upon the action of the governor. Air in excess of that required to burn the fuel is introduced into the cylinder, and accordingly perfect combustion takes place The speed of the engine is controlled by means of the governor acting on the by-pass valves (one for each fuel pump). The by-pass oil valves are opened by arms pivoted on a shaft raised or lowered by the governor, and operate as follows: If only a small amount of fuel is required in the cylinder to overcome the load, the governor holds the by-pass valve open for a lengthened period and a greater amount of the oil is allowed to return to the suction pipe, while, if the load is greater, and consequently more fuel is required in the cylinder to overcome it, the by-pass valves open for a relatively shorter period and then less oil returns to the suction pipe, a greater amount of fuel passing to the cylinder. By this method of governing a very close regulation of speed is effected.

Indicator card from this engine is shown at Fig. 100.

The Diesel engine has created great interest in engineering circles the world over, and many tests have been made of it. Professor Denton, of the Stevens Institute, Hoboken, N. J., in 1898 conducted

a series of tests on this engine, and according to his report of those tests the consumption of fuel was 0.534 lbs. per B. H. P. per hour at full load, and at less than half load 0.72 lbs. per B. H. P. per hour. This is

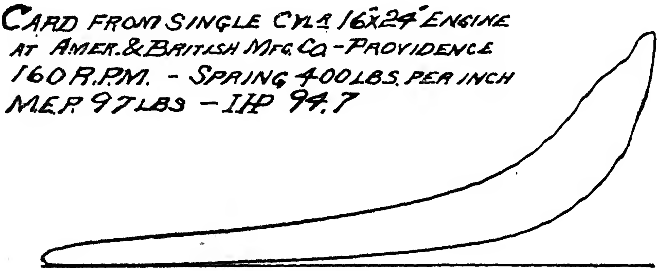


FIG. 100.

equivalent to a thermal efficiency (on the I. H. P.) of 37.7 per cent.

The following is the heat-balance table as shown by Professor Denton:

	PER CENT.
Heat of combustion accounted for by indicated power	37.2
Removed by jacket	35.4
Remainder	27.4
Total heat of combustion	100.0

Another type of the Diesel engine, that made by the manufacturers in Sweden, is shown at Fig. 101.

The following tests were made by Prof. Meyer in 1900 on a German type 30 H. P. engine. The

cylinder 11.8″ diam., 18.1″ stroke, air-pump cylinder 1.9″ diam., 3.1″ stroke. Air was taken from motor cylinder at a pressure of 20 atmospheres and com-

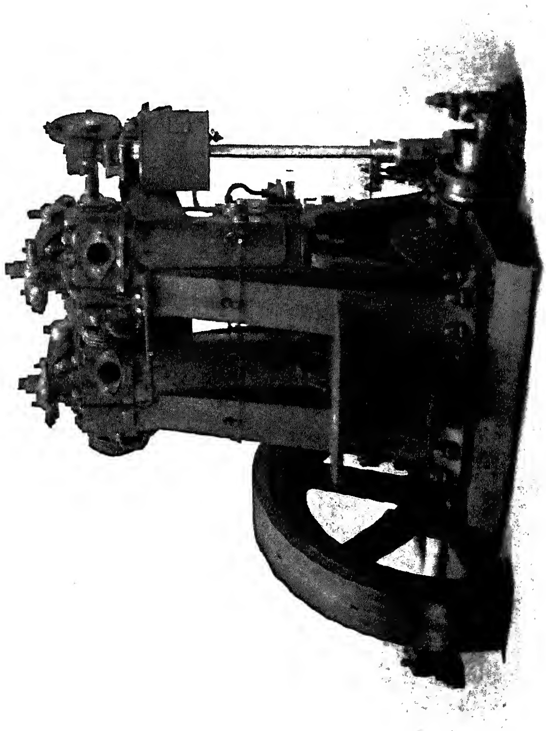


FIG. 101.

pressed to 45 or 60 atmospheres. Negative work in the motor cylinder was equivalent to 5.66 H. P. at 181.

R. P. M. The air pump was not indicated, consequently the effective power is not given. The mean indicated pressure at normal load was approximately 90 lbs. per square inch. The exhaust gases were invisible. Two kinds of fuel were used, American petroleum, specific gravity 0.79, having 18,540 B. T. U. per lb., and Tegern See (Bavaria) crude oil, specific gravity 0.789.*

TABLE VIII.—RESULTS OF TRIALS OF A DIESEL OIL ENGINE (MEYER), 1900.

	American Petroleum.				Raw Tegern See Oil.		
Load on Brake.	Full Load.	Nor-mal.	¾ Load.	Half Load.	Nor-mal.	¾ Load.	Half Load.
Revs. per minute....	177.4	181.1	184.0	183.3	181.2	181.8	185.0
Brake (or actual) H. P., metric......	39.45	30.17	23.81	15.26	30.18	23.5	15.4
Indicated H. P. (motor cyl.)...........	48.2	39.52	33.10	25.02	40.96	33.0	26.4
Mech. efficiency.....	82	76	72	61	73	71	58
Oil used per B. H. P., per hour.......lbs.	0.48	0.45	0.48	0.57	0.47	0.49	0.57
Percentage of heat of oil as useful work............	28	30	28	24	29.8	.....	......

CRUDE OIL VAPORIZER.

On the Pacific Coast crude oil is now being largely used for fuel. In many instances this fuel is used, being vaporized or gasified in a separate apparatus and is then consumed in the ordinary gas engine. This

*"Gas and Petroleum Engines." By Prof. Wm. Robinson. Second edition. Page 777.

apparatus is separate from the engine, the oil being entirely gasified before it reaches the engine cylinder. Such vaporizing apparatus or retorts are made by various manufacturers, but in general principle they are similar. The heat of the exhaust gases from the engine is utilized to heat the retort into which the oil is introduced, where it is gasified.

Mr. Frank H. Bates has drawn attention to these various retorts, which usually consist of a cast-iron chamber enclosing an inner chamber, also of cast iron.* The fuel to be gasified enters the inner ribbed chamber through suitable openings, and the gas is drawn from the chamber through a separate connection from the inner chamber to the engine cylinder. The exhaust gases from the engine are connected to the outer chamber and pass around, heating the inner chamber to a temperature necessary for vaporization. Provision is made to draw off the residue of the crude oil, which is not capable of vaporization, and provision is also made to cleanse the vaporizing chamber of deposit of carbon and other solid matter.

In the "Economist" retort the inner ribbed chamber, or drum, is made to slowly revolve, and, the ribs being spirally shaped, the oil is propelled from end to end and the heat is then equally distributed around the inner chamber. In service where the load is fairly constant, and where opportunity to cleanse the retort occasionally, is afforded, these retorts have given excellent results. For installations, however, such as

*See *Journal of Electricity, Power and Gas,* Vol. XIII., p. 5.

electric railway service, or where the load varies between wide limits and where continuous running is imperative, it is stated that difficulty has been experi-

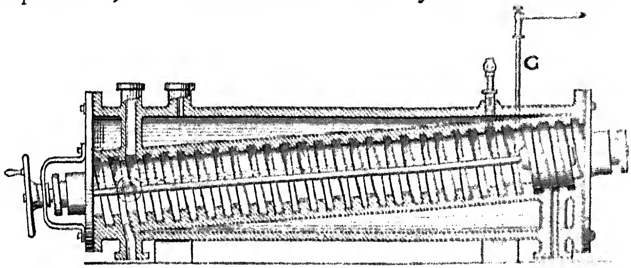


FIG. 102.

enced, due to the fluctuating temperature of the retort heated by the exhaust gases, which involves improperly regulated supply of vapor to the cylinder. To overcome this difficulty with varying loads, Messrs. C. C. Moore & Co. have developed an improved system of using crude oil in connection with gas engines.

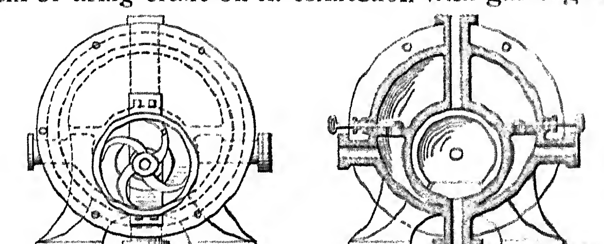


FIG. 103.

The generator, as made by this company, is shown in Figs. 102 and 103, in which are shown a longitudinal elevation of the generator, end elevation, and also the

generator connected up to its drainage chamber for the automatic removal of the deposit. It will be noted from Fig. 102 that a scraper is arranged which can be moved from end to end of the vaporizer by means of the hand wheel. This scraper is shown in Fig. 105. The oil supply is regulated by means of a thermostatic valve, and is automatically maintained at a constant level by this means. The method of operation is as follows:

Oil is first fed into the vaporizing chamber by means of a valve until the level in both this chamber and in the oil feed device is a little above the level of the upper drain pipe. A heating device is then inserted into the exhaust gas passage, heating the vaporizing chamber to about 300° Fahr. The engine is started by means of compressed air, and when in operation air heavily charged with oil vapor passes through the nozzle *G*, Fig. 102, to the engine cylinder. The exhaust gases from the engine afterwards furnish the heat necessary to maintain the vaporizer at a proper temperature; these gases pass around the generator, and thence by the exhaust pipe to the roof. The temperature of this chamber is regulated by the thermostatic valve, which, when the temperature of the vaporizer rises too high, allows the exhaust gases to be by-passed from the vaporizer and pass directly to the roof. The thermostatic device consists of an aluminum tube inserted directly into the vapor chamber, around which the exhaust gases pass. The aluminum tube is closed at its upper end and is attached to a system of levers so arranged as to exaggerate its move-

ment, caused by the variation in temperature. Accordingly, when the temperature of the vaporizer chamber rises above that required, the expansion of the aluminum tube is arranged to close a needle valve, which allows the pressure of the exhaust gases from the engine to lift a larger valve, thus opening a passage outside the vaporizer, through which the exhaust passes instead of entering the chamber around the vaporizing retort. By this means the temperature of the retort is regulated within very close limits.

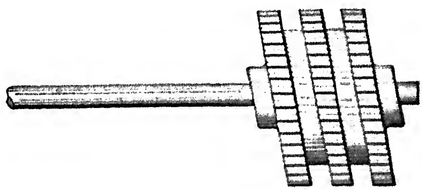
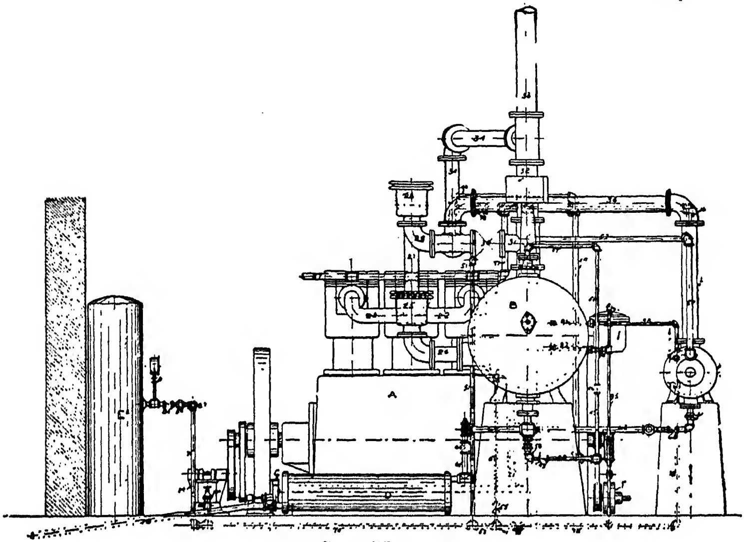


FIG. 105.

The proper level of the liquid fuel to be vaporized is regulated by an automatic ball check valve placed in the chamber marked *I*, Fig. 106, through which the oil passes to the vaporizer. A relief valve is inserted in the supply pump, so that when the valve to the vaporizing chamber is closed the fuel can by this means flow back to the tank. The retort is readily cleansed by means of the scraper already referred to, shown in Fig. 105, which is operated by hand periodically. In the larger size installations made by Messrs. C. C. Moore & Co. more extensive equipment is provided, in which arrangement is made to utilize the heat rejected by the exhaust gases and also



SIDE VIEW

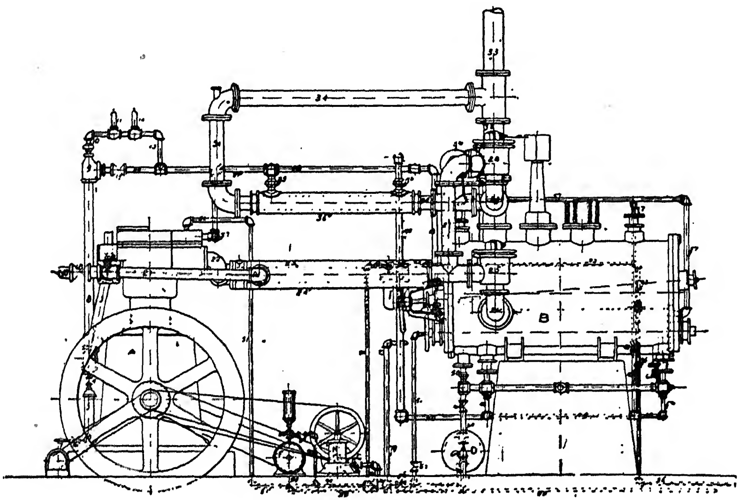


FIG. 106.—END VIEW.

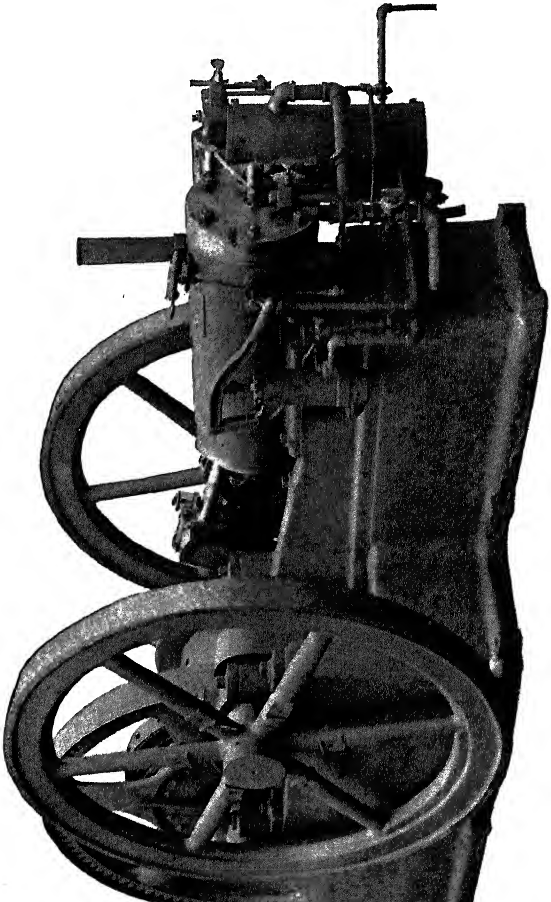


Fig. 108.

the heat given off from the water jacket, and in which installations the residue of the oil is partly used also. In these outfits a combination of oil vapor and water gas is formed, two superheaters being added, one of which is heated by the exhaust gases, in which part of the cooling water issuing from the water jacket is turned into steam; the second superheater is heated by the burning of residue oil in connection with compressed air. In this way, it is stated, steam raised to approximately 1600° Fahr. in the chamber *C*, Fig. 106, is mingled with the oil vapor forming the combination of oil vapor and water gas referred to. By the use of this apparatus a greater economy is effected and a greater part of the heat of the fuel utilized.

The following is a brief description of the accompanying illustrations, Fig. 106:

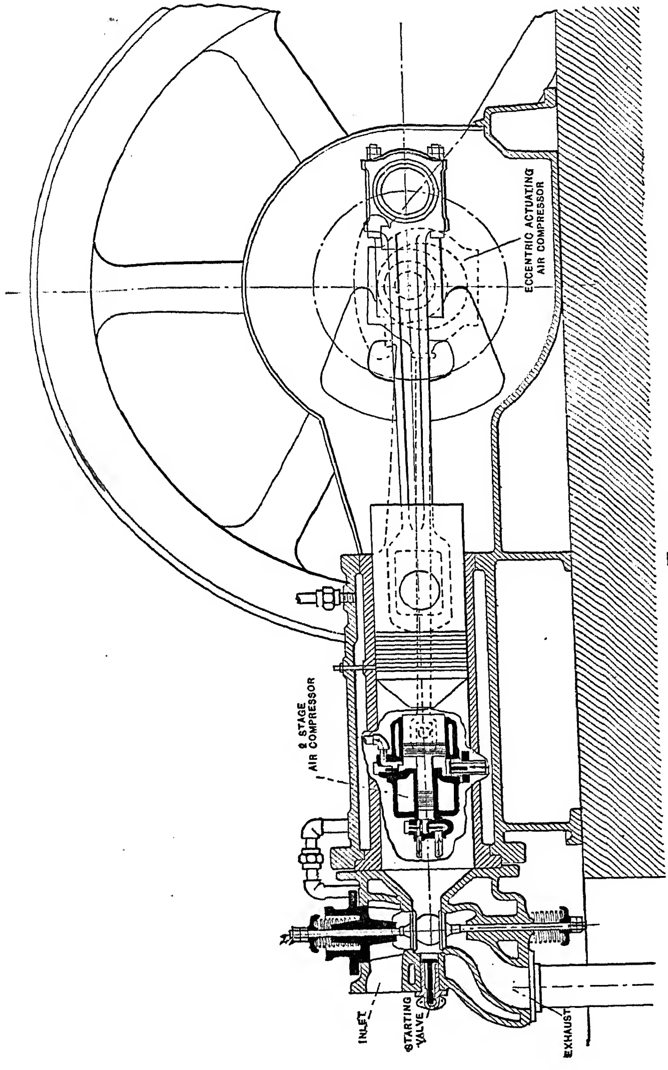
The three-cylinder Westinghouse gas engine of the vertical type is shown at *A*. The generator by which the crude oil is vaporized is shown at *B*. The superheater (heated by residual oil burners) is marked *C*. The chamber for drainage of residuals is shown at *D*. *H* is an air-compressor operated by belt from the engine crank-shaft. *I* is the automatic oil feed, which maintains the proper level of the oil in the generator. E, E^1 and E^2 are the air storage tanks maintained at a pressure of 160 lbs. per square inch. *F* is the rotary oil pump which raises the fuel from the storage tank underground to the vaporizer. The water-circulating pump which supplies the cooling water to the cylinders is shown at *G*.

A separate vaporizing attachment for using crude

oil of the type already mentioned is shown at Fig. 108. The vaporizer is separate from the engine, being attached to the gas or gasoline engine, where it is required to use crude oil as fuel instead of gas or gasoline. The outfit shown is the Fairbanks-Morse gas or gasoline engine, which has attached to it the outside apparatus for vaporizing the oil, the vaporizer being a cast-iron chamber into which the liquid oil is injected. This chamber is heated while in operation by the exhaust gases. Before starting it is necessary to use an outside lamp, in order that the chamber may become heated to the temperature required to vaporize the fuel. The oil is mixed with air drawn into the vaporizer and becomes vaporized in this chamber, and is drawn therefrom into the cylinder in the usual way.

As will be seen from the illustration, the engine shown at Fig. 108 is geared directly up to hoisting drum. These outfits are very largely used for mining and similar purposes, where hoisting engines can be readily utilized.

A new type of oil engine, made in sizes from 85 H. P. upwards, is shown at Fig. 109. This engine is manufactured and patented by the De La Vergne Machine Company and is known as their Type FH oil engine. It operates on the four-cycle principle, and is single acting, of the horizontal type, and is furnished in either single or twin cylinder units. The largest size which this company has furnished hitherto is 250 H. P. twin cylinder, but engines of larger size are in course of construction.



ECCENTRIC ACTUATING
AIR COMPRESSOR
2 STAGE
AIR COMPRESSOR
INLET
STARTING
VALVE
EXHAUST

FIG. 109.

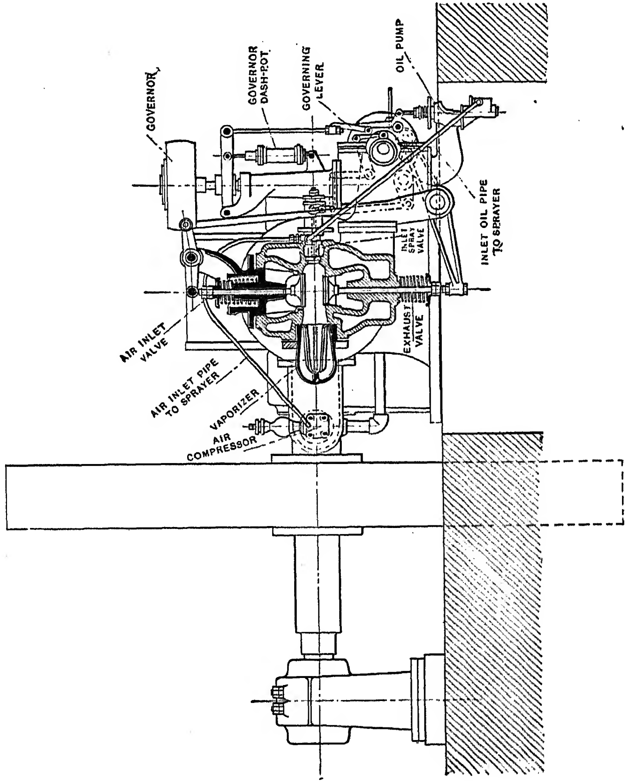

GOVERNOR
GOVERNOR DASH-POT
GOVERNING LEVER
OIL PUMP
INLET OIL PIPE TO SPRAYER
INLET SPRAY VALVE
EXHAUST VALVE
AIR INLET VALVE
AIR INLET PIPE TO SPRAYER
VAPORIZER
AIR COMPRESSOR

FIG. 110.

This engine is equipped with a two-stage air compressor shown in the sectional view at Fig. 109, which is operated directly from the crank-shaft by an eccentric. The compressed air is used for spraying purposes and is injected into the vaporizer and combustion space with the fuel, thus insuring complete spraying of the fuel as it enters the vaporizer. Briefly stated, the method of operation of this engine is as follows:

At the first stroke of the piston outwards, air is drawn into the cylinder through an inlet valve on the top of the breech end or valve chamber. On the second or inward stroke of the piston, compression takes place. As will be seen from the indicator card at Fig. 112 the maximum pressure of compression is 260 lbs. As the process of compression is completed the fuel (fuel or crude oil as heavy as 14° Beaume) is injected into the vaporizer and mingles with the compressed air already referred to.

The spray valve shown in section Fig. *7a* is positively controlled by an independent cam on the cam-shaft. The compressed air furnished by the two-stage air compressor is delivered at the sprayer at about 400 lbs. pressure. Only a small amount of air (about 2% of the cylinder volume) is delivered at each injection. Immediately the fuel enters the combustion space and comes in contact with the air heated by the process of compression together with the heated walls of the vaporizing chamber ignition takes place, and on the third or outward stroke of the piston expansion begins. The maximum pressure, as will be seen from the indicator card, is slightly over 400 lbs. At a point 85% of

the stroke, the exhaust valve is opened, allowing the products of combustion to escape.

The vaporizer of this engine is a rough gun-iron casting, somewhat similar to that of Type 2 described on page 8, but without contracted opening. The oil pump is operated from the cam-shaft and has the length of its stroke varied by the governor in accordance with the load requirements.

LOAD FACTOR	0	¼	½	¾	1	
DATE	MAY 24-09	MAY 25-09	MAY 25-09	MAY 25-09	MAY 24-09	MAY 24-09
LENGHT OF TEST	1 HOUR	½ HOUR	½ HOUR	¾ HOUR	½ HOUR	20'
K.W		24.5	48,4	68,9	95	108,8
EFFICIENCY % OF GENER. & BELT		76	83.5	85	85.5	86
BHP		43.8	78.6	110	151	171.8
OIL USED LBS	13.38	10934	17.0625	23	29,75	22,90
OIL PER HOUR LBS	13.38	21,87	34,125	44,6	59,5	68,72
OIL PER KW HOUR LBS		0,89	0,71	0,647	0,625	0,630
OIL PER BHP.		0,498	0,437	0,405	0,393	0,398

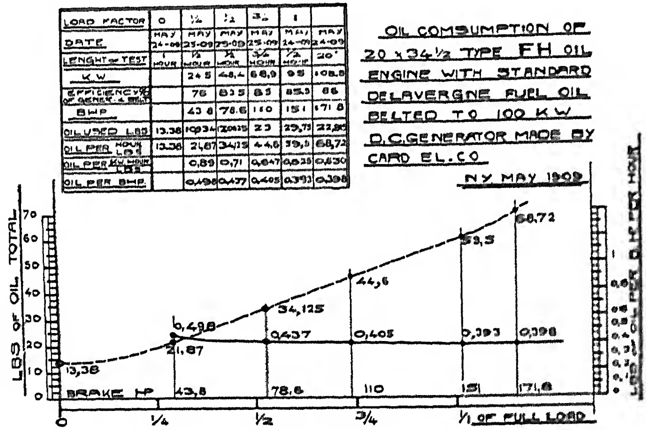


FIG. 111.

This engine is of the best design in every detail and of very heavy construction. The marked economy is shown by diagram, Fig. 111, from which it will be seen that a fuel consumption as low as 0.393 lb. of crude oil per actual horse-power per hour has been obtained.

Tests have also shown the fuel economy to be as low as 0.437 lb. at half load. (See page 248.)

The cams operating the air and exhaust valves are

accurately designed and machined. The engine is almost silent in operation. The starting is effected in the ordinary way by means of compressed air, as explained on page 105. The vaporizing chamber is heated for a few minutes before starting by means of an external lamp in a similar way as with Type 2 engines (page 8).

The regulation of speed is effected by a Hartung governor operated by gears from the cam-shaft, which actuates through levers directly on the oil supply pump, lengthening or shortening the stroke in accordance with the requirements of the load.

At this time only a few installations of this engine have been made, but the makers state that under continued and exhaustive tests made by independent engineers results even better than those shown in the accompanying diagram have been obtained.

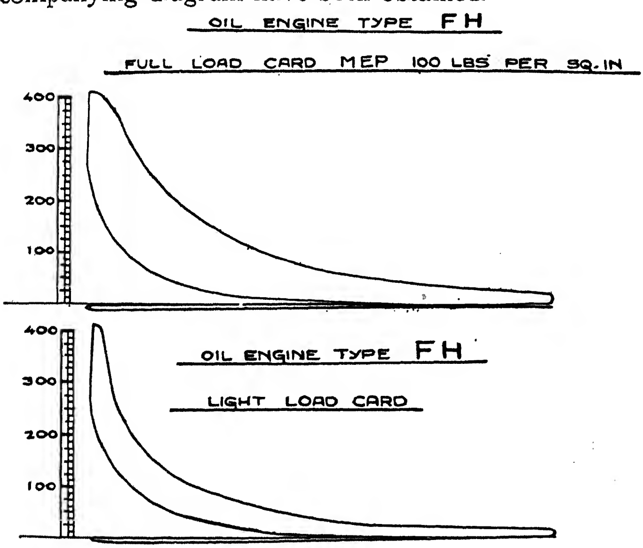


FIG. 112.

CHAPTER XIII.

FUELS.

The fuel to be used in the type of engines here discussed is frequently a matter of inquiry, and accordingly a brief description of the various fuels used is given.

The Texas oil, which hitherto has not been so fully treated of elsewhere is discussed more fully than the other fuels.

The supply of petroleum is produced chiefly in the United States of America and in Russia, while it is also found in many other countries in small quantities.

Petroleum is found in the United States in the Central Eastern States, notably Pennsylvania, New York, Ohio and West Virginia; in Texas in the region around Beaumont and Corsicana, in California chiefly in the Kern County, Coalinga, Los Angeles, producing fields. In Russia oil fields are found around Baku and in the range of the Caucasus Mountains.

Paraffin or shale oil, a fuel produced by a slow process of distillation of "shale" and bituminous coal, is also produced in Scotland.

Crude petroleum as it issues or is pumped from the earth contains a variety of hydrocarbons of different characteristics, and after its sediment has settled it is

subjected to a process of refining known as fractional distillation, by which process the various hydrocarbons are separated and are afterwards condensed into the different products known in commerce as benzine, gasoline, naphtha, being the lighter products, having a flash-point below 73° Fahr. Next the illuminating oils, such as W. W. 150° kerosene, White Rose and other brands of a similar composition, are obtained, having a flash-point above 73° Fahr. The next product is gas oil, or fuel oil, used largely for gas-making and also as fuel in internal combustion engines, having a flash-point of about 190°. Lubricating oils, paraffin, wax, vaseline, etc., are afterwards procured, the residue being only a heavy liquid sometimes used for fuel.

The fuels used chiefly in the engines here discussed, as already stated, are the crude oils, the illuminating oils and the fuel or gas oil.

Crude Oils.

In the accompanying tables will be found the characteristics of the crude oils produced from the different Russian oil fields, the American oil fields of the Allegheny region, as well as the oils produced in Texas, California and elsewhere.

The Russian crude oil is heavier than the American product found in the Allegheny region, the average specific gravity of the former being .85, that of the latter being .79.

Texas crude oil, many samples of which have been used by the writer in the Hornsby-Akroyd oil engine,

is a dark, heavy liquid having a specific gravity varying from .861 to .915, the flash-point (open method) being 180° to 195°.

An analysis of this oil by Messrs. Clifford Richardson and E. C. Wallace,* taken from the Lucas well, Beaumont, Texas, 1901, in which the following, it may be mentioned, were the methods of examination, has been made.

The specific gravity was determined in a picnometer at 25° C., the flash-point was taken in a New York State oil tester, the refractive index with an Abbe refractometer at 25° C. The viscosity represents the number of seconds required for the oils to flow from a 100 c.c. pipette, according to the P. R. R. specifications. Volatility was obtained by allowing 20 grm. of crude petroleum to be heated in an open dish $2\frac{1}{2}$ inches diameter, $1\frac{1}{4}$ inches deep, to various temperatures for various periods of time, or until the loss became small enough to neglect. The volatilization then goes on below the boiling point. The vapor not being confined, there is no "cracking." The distillation in Engler's Flask was carried out in the usual way, the distillate between 150° and 300° C. representing the burning oil available commercially.

For the purpose of fractional distillation, about half a litre of oil was distilled in a litre flask of the Engler shape (but larger) supported on a six-mesh iron cloth surrounded by loose bricks covered with asbestos board. The distillate was condensed in an air-con-

*See "Journal of the Society of Chemical Industry," Vol. 20, No. 7.

denser 3 feet long connected with a Bruhl's receiver, where a vacuum of 20 mm. could be maintained. All joints were mercury sealed or of solid glass; access of air or decomposition was prevented. A current of carbon dioxide was conducted to the bottom of the distilling flask to agitate the oil and remove air from the apparatus. The oil was heated by a ring-flame Fletcher burner, and distilled at ordinary pressure as long as there were no signs of cracking. As soon as any decomposition was recognized, or the temperature had reached a high figure, the oil was cooled and the vacuum made. The difference in boiling point at atmospheric pressure and at 20 mm. for hydrocarbons, boiling under 760 mm. at about 320° C., is 117°, a distillate coming over at 317° at atmospheric pressure beginning to distil at 200° in a vacuum of 20 mm. The distillates were then treated twice with an excess of sulphuric acid, washed with dilute soda, dried over sodium, and then determinations repeated. Finally, one of the distillates was treated with a mixture of equal volumes of sulphuric and nitric acid, washed, boiled with sodium and examined.

Examination of Residues.—The residues left after evaporation in the open dish, or from either of the methods of distillation, are characteristic and of value in determining the nature of any petroleum, and as to whether it has a so-called asphaltic or paraffin base.

Ultimate Analyses.—These were made with the precautions which have been found necessary in burning the polymethylene hydrocarbons, which very readily escape complete combustion.

Beaumont oil contains a much larger proportion of unsaturated hydrocarbons removable by sulphuric acid than either Pennsylvania or Ohio petroleum. The Beaumont oil has a high sulphur content and carries, as it comes from the wells, a large amount of hydrogen sulphide in solution. This gas has previously been observed in solution in petroleum, but not in so large quantity as at Beaumont. The sulphuretted hydrogen is largely lost on standing, and more completely on blowing air through it. After such treatment the oil contained 1.75 per cent. of sulphur in the form of sulphur derivatives of the hydrocarbons.

A comparison of the ultimate compositions of the Texas oil with other oils used for fuel shows that, while not equal to Pennsylvania and Ohio oils, owing to the low carbon and high sulphur, it is not inferior to the California petroleums in any marked degree.

TABLE IX.—ULTIMATE COMPOSITION.

	Beaumont.	Penna.*	Ohio †
Carbon....	85.03	86.10	85.00
Hydrogen........................	12.30	13.90	13.80
Sulphur	1.75	0.06	0.60
Oxygen and Hydrogen..........	0.92	.....	0.60
Loss on treatment with excess of H_2SO_4. (Sulphuric acid).... ..	39.0	21.0	30.0

* Engler. † Mabery, Noble Co.

TABLE X.—BEAUMONT OIL.

Specific gravity 25° C......	0.912	0.914	0.8014	0.8293
Flash..........................	Ord. Temp.	110°	Ord.	Ord.
Viscosity, P.R.R. pipette..	......	75"	42"	37"

TABLE XI.—VOLATILITY IN OPEN DISH.

	Per Cent.	Per Cent.	Per Cent.	Per Cent.
110° C., 230° F.: 7 hours....	19.19	20.0	41.2	47.3
162° C., 325° F. 7 "	31.31	27.0	43.0	58.0
205° C., 400° F. 7 "	57.57	49.0	59.0	68.0
To constant weight—				
105° C., 221° F.: 42 hours	48.0	48.0	48.7	58.7
162° C., 325° F.: 70 "	64.0*	57.0	61.0	71.8†
205° C., 400° F.: 49 "	74.0	74.0	75.0	84.0

*49 hours. †42 hours.

TABLE XII.—DISTILLATION: ENGLER'S FLASKS.

	Beaumont.	Ohio.	Pennsylvania.
Distillation begins..............	110° C.	85° C.	80° C.
Below 150° C..........per cent.	2.5	23.0	21.0
150°—300° C. .... "	40.0	21.0	41.0
300°—350° C. .... "	20.0	21.0	14.0
350°—400° C. .... "	25.0	27.0	{ 23.0 { 99.0
Loss on acid treatment (150°—300° C. fraction)............	10.0	5.0	1.8
150°—260° C..........per cent.	30.0	....	....
Loss on acid treatment. "	8.0	....	....
Percentage of acid used "	7.0	2.5	2.0

TABLE XIII.—SPECIFIC GRAVITY AND REFRACTIVE INDEX.

	Beaumont.		Ohio.		Pennsylvania.	
	Sp. Gr.	Refrac. Index.	Sp. Gr.	Refrac. Index.	Sp. Gr.	Refrac. Index.
Below 150°..	(Amount too small.)		0.7297	1.412	0.7188	1.415
150°—300°..	0.8749	1.473	0.8014	1.442	0.7984	1.437
300°—350°..	0.9089	1.501	0.8404	1.468	0.8338	1.462
350°—400°..	0.9182	1.508	0.8643	1.481	Paraffin	1.470
	After acid treatment.					
150°—300°..	0.8704	1.473	0.8006	1.443	0.7791	1.438

Table XIV.—Calorific Power of Various Descriptions of Petroleum, Etc. (B. Redwood.)

Description of Oil.	Specific Gravity at 0° C.	Chemical Composition. Carbon.	Hydrogen.	Oxygen.	Coefficient of Expansion.	Am't of Water Evaporated Per Unit of Fuel.	Effect in Heat Units.
Heavy Petroleum from West Virginia.......	0.873	83.5	13.3	3.2	0.00072	14.58	10,180
Light Petroleum from West Virginia.......	0.8412	84.3	14.1	1.6	0.000839	14.55	10,223
Light Petroleum from Pennsylvania........	0.816	82.0	14.8	3.2	0.00084	14.05	9,963
Heavy Petroleum from Pennsylvania........	0.886	84.9	13.7	1.04	0.000721	15.30	10,672
American Petroleum..	0.820	83.4	14.7	1.9	0.000868	14.14	9,771
Petroleum from Parma	0.786	84.0	13.4	1.8	0.000706	13.96	10,121
Petroleum from Pechelbronn	0.912	86.9	11.8	1.3	0.000767	14.30	9,708
Petroleum from Pechelbronn............	0.892	85.7	12.0	2.3	0.000793	14.48	10,020
Petroleum from Schwabweiler	0.861	86.2	13.3	0.5	0.000858	15.36	10,458
Petroleum from Schwabweiler	0.829	79.5	13.6	6.9	0.000843	.....	......
Petroleum from Hanover	0.892	80.4	12.7	6.9	0.000772	.....	......
Petroleum from Hanover	0.955	86.2	11.4	2.4	0.000641	.....	......
Petroleum from East Galicia..............	0.870	82.2	12.1	5.7	0.000813	14.23	10,085
Petroleum from West Galicia...............	0.885	85.3	12.6	2.1 (N. O.)	0.000775	14.79	10,231
Shale Oil from Ardèche	0.911	80.3	11.5	8.2 (O. S. N.)	0.000896	12.24	9,046
Coal Tar from Paris Gasworks	1.044	82.0	7.6	10.4	0.000743	12.77	8,916
Petroleum from Balakhany.................	0.822	87.4	12.5	0.1	0.000817	.....	11,700
Light Petroleum from Baku................	0.844	86.3	13.6	0.1	0.000724	16.40	11,460
Heavy Petroleum from Baku................	0.938	86.6	12.3	1.1	0.000681	15.55	10,800
Petroleum residues from Baku Factories	0.928	87.1	11.7	1.2	0.00091	.....	10,700
Petroleum from Java..	0.923	87.1	12.0	0.9	0.000769	15.02	10,831
Heavy Oil from Ogaio	0.985	87.1	10.4	2.5	0.0008685	14.75	10,081

TABLE XV.—COMPOSITION, PHYSICAL PROPERTIES, ETC., OF VARIOUS DESCRIPTIONS OF PETROLEUM.

(B. REDWOOD.)

Description of Petroleum, etc	Elementary Composition			Percentage of Distillate at °C.										Specific Gravity at °C.				Coefficient of Expansion.	Composition of Distillate.			Specific Gravity of Distillate at °C.		Specific Gravity of Residue at °C.		Calorific Power.
	C	H	O	100°	120°	140°	160°	180°	200°	220°	240°	260°	280°						C	H	O					
Heavy Virginia Petroleum (135 m.)...........	83.5	13.3	3.2	1.0	....	1.3	....	12.0	....	....	....	....	....	0°	0.873	50.1°	0.853	0.00072	85.3	13.9	0.8	13°	0.819	13.3°	0.864	10,180
Light Virginia Petroleum (200 m.)...........	84.3	14.1	1.6	1.3	4.3	11.0	17.7	25.[illegible]	28.5	....	....	....	....	0°	0.8412	50.1°	0.808	0.000830	84.0	14.4	1.6	14°	0.762	14°	0.860	10,2[illegible]3
Light Pennsylvania Petroleum (200 m.)....	82.0	14.8	3.2	4.3	10.7	16.0	23.7	28.7	31.0	....	....	....	....	0°	0.8·6	50.1°	0.784	0.00084	85.1	14.3	0.6	13.6°	0.735	13.6°	0.845	9,963
Heavy Ohio Petroleum.......	84.2	13 1	2.7	...	....	....	....	....	....	....	....	....	....	0°	0.887	53°	0.85[illegible]	0.000748	85.4	14.0	0.6	.....	.....	14.8°	0.860	10,399
Heavy Pennsylvania Petroleum (200 m.)....	84.9	13.7	1.4	...	....	....	....	....	...	....	....	....	12.0	0°	0.886	50.1°	0.853	0.000721	At 350° 86.7 85.4	12.2 13.8	1.1 0.8	14° 13.2°	0.762 0.802	13°	0.875	10,672
Java Petroleum	87.1	12.0	0.9	1.0	1.0	....	....	7.7	15.0	22.3	24.3	..	....	0°	0.923	53°	0.888	0.000764	86.2	12.2	1.6	.....	0.811	13.3°	0.935	10,831
" "	83.6	14.0	2.14	0.8	3.0	9.3	16.3	22.0	27.8	....	....	....	....	0°	0.827	53°	0.78	0.000923	83.9	14.1	2.0	13.1°	0.778	13.3°	0.914	9,593
" "	85.0	11.2	2.8	...	....	....	....	. ..	....	2.3	4.0	9.0	....	0°	0.972	53°	0.945	0.000652	85.1	12.2	1.7	13.2°	0.762	13.2°	0.942	10,183
East Galicia Petroleum.......	82.2	12.1	5.7	2.1	4.6	8.7	13.7	14.3	21.7	25.3	32.3	....	....	0°	0.870	50°	0.83[illegible]	0.000813	80.5	13.6	5.9	21°	0.778	21°	0.901	10,005
West Galicia Petroleum.......	85.3	12.6	2.1	...	4.0	9.8	14.3	23.3	27.0	30.7	36.7	....	....	0°	0.885	.....	0.852	0.000775	83.8	12.9	3.3	21°	0.786	21°	0.931	10,231

TABLE XVI.—OIL FUEL. (B. REDWOOD.)

Locality.	Fuel.	Sp. Gr. at 0° C.	Chemical Composition.			Heating Power.	
			Carbon.	Hydrogen.	Oxygen.	Actual Calorimetric (lb. C. Units.)	Calculated (lb. C. Heat Units.)
Russian............	Petrol. refuse	0.928	87.1	11.7	1.2	......	11,018
"	Astatki	0.9	84.94	13.96	1.2	10,340	11,626
Caucasian..........	Heavy Crude	0.938	86.6	12.3	1.1	10,800	11,200
" (Novorossisk)	" "	.....	84.9	11.63	1.458	10,328	......
Pennsylvanian......	" "	0.886	84.9	13.7	1.4	......	10,672
American..........	" "	.....	86.894	13.107	.....	10,912	......
"	Refined	.....	85.491	14.216	0.293	11,045	......
"	Double "	.....	80.583	15.101	4.316	11,086	......
"	Crude "	.....	83.012	13.889	3.099	11,094	......

TABLE XVII.—CALORIFIC POWER OF CRUDE PETROLEUM. (B. REDWOOD.)

	Sp. Gr.	Calories.
Heavy Lubricating Oil, White Oak, Western Virginia	0.873	10,180
Light Illuminating Oil, Oil Creek, Pa.	0.816	9,963
Oil from Dandang, Leo Rembang, Java.	0.923	10,831
Light Oil from Baku..................	0.884	11,460
Oil from Western Galicia............	0.885	10,231
" " Eastern "	0.870	10,005
" " Parma	0.786	10,121
" " Schwabweiler	0.861	10,458

California Crude Oil.

The crude petroleum procured in the various oil fields of California, from the information available, appears to vary considerably in its characteristics. According to the report of the Chamber of Commerce of San Francisco, in 1902 the oil-producing fields of Kern River, Coalinga, Los Angeles, Fullerton, with many others, in which over 2,000 wells were in operation, produced an average daily supply of over 37,000 barrels. It has been used hitherto chiefly for fuel purposes, and having in most instances an asphaltum base, is most suitable for this purpose. The characteristics of the oil vary so widely, however, that while some samples can only be used for fuel, that produced in other wells would yield illuminating oils on distillation in considerable quantity. The following is the analysis of two samples of the distillates from the Kern River field:

		(Flash test was taken, using the open method.)
Gravity	..........0.901	0.859
Beaumé	..........26.2°	34°
Flash	............169° F.	119° F.

According to Mr. Paul Prutzman,* the oil produced in Coalinga oil field varies from 11.5° Béaume to 45°. The viscosity of various samples varies from 68 to 296, while the flash point varies from 220° to 278° F. This writer also refers to the refining qualities of various samples, from which it would appear that on distillation

* *Pacific Oil Reporter,* Vol. 4, No. 35.

while some of the oil would give far greater amount of kerosene (42° B.) than others, the average yield of kerosene on distillation would be about 14 per cent; while the engine distillate (48 to 52° B.) given off from the above-mentioned samples would also vary considerably in quantity, the average would, however, be approximately 14 per cent—the products which were obtained being of a lighter quality than kerosene were inconsiderable. This fuel is now used on the Pacific coast in large quantities, both under boilers for generating steam, in gas engines having first been gasified, as explained in Chapter XII., as well as in the oil engine proper, where it is vaporized by the same methods as with kerosene.

FUEL OIL.

The oil known as fuel or gas oil, as already stated, is procured in the process of fractional distillation after the lighter oils and the illuminating oils have been taken off. Various samples of this fuel have come within the writer's notice, the characteristics of which have varied considerably, as will be seen from the following table:

FUEL OIL.

Specific gravity ..	0.832	.878
Beaumé	36°	30.2°
Flash-point	144° F.	298° F.
Fire test	183° F.	247° F.

This fuel is much used in oil engines in the United States. With the heavier grades a slight deposit of carbon is left in the engines, which requires periodical removing.

TABLE—THE CALORIFIC POWER OF PETROLEUM OILS AND THE RELATION OF DENSITY TO CALORIFIC POWER.

The following are extracts of tests of various samples of crude oils, representing the products from the principal oil fields of the United States, and were made by H. C. Sherman and A. H. Kropf, at Columbia University, N. Y., during 1908, and are reprinted from the Journal of the American Chemical Society.*

DENSITIES AND HEATS OF COMBUSTION OBSERVED AND CALCULATED.

Specific Gravity, 15°/15°.	Baume Degrees	Calories per Gram.	B. T. U. per Pound.	B. T. U. calculated.	Percentage Error.	Description.
0.7100	67.2	11,733	21,120	20,938	— 0.91	Gasoline.
0.7830	48.8	11,121	20,018	20,206	+ 0.92	Kerosene.
0.7850	48.35	11,119	20,014	20,194	+ 0.89	Cal. refined.
0.7945	46.2	11,128	20,030	20,098	+ 0.33	W. Va. crude.
0.8048	44.0	11,149	20,068	20,010	— 0.29	Ohio crude.
0.8059	43.7	11,143	20,057	19,998	— 0 29	Penna. crude.
0.8080	43.2	11,001	19.802	19,979	+ 0.88	Cal. refined.
0.8103	42.8	11,090	19,963	19,962	± 0.00	Kansas refined.
0.8237	40.0	10,981	19,766	19,850	+ 0.42	W. Va. crude.
0.8324	38.2	10,990	19,782	19,778	— 0.02	Penna. crude.
0.8418	36.3	10,950	19,710	19,702	— 0.04	Ohio crude.
0.8421	36.25	10,997	19,795	19,698	— 0.48	Indian Ter.
0.8436	36.0	11,069	19,924	19,690	— 1.17	Indian Ter.
0.8510	34.5	10,958	19,724	19,630	— 0.47	Kansas crude.
0.8580	33.2	10,772	19.389	19,578	+ 0.95	Kansas crude.
0.8597	32.8	10,766	19,379	19,562	+ 0.95	Illinois crude.
0.8640	32.05	10,867	19,555	19,530	— 0.12	California Ref.
0.8914	27.1	10,690	19,242	19.332	+ 0.45	Texas crude.
0.8970	26.1	10,753	19,355	19,294	— 0.31	Texas crude.
0.9065	24.45	10,751	19,352	19,228	— 0.63	Texas crude.
0.9087	24.1	10,712	19,282	19,213	— 0.35	Texas crude.
0.9158	22.9	10,318	18,572	19,166	+ 2.58	Calif. crude.
0.9170	22.7	10,613	19,103	19,157	+ 0.28	Fuel oil.
0.9644	15.2	10,327	18,589	18,858	+ 1.42	Calif. crude.

* Journal American Chemical Society, Vol. XXX, No. 10, October, 1908.

CHAPTER XIV.

MISCELLANEOUS.

OWING to the increasing use of the metric system, the following comparisons of United States and metric measures and weights, etc., prepared by C. H. Herter, are added. The unit of length is the metre = 39.37 inches; the unit of capacity is the litre = 61.0236 cubic inches; the unit of weight is the gramme = 15.43236 grains.

The following prefixes are used for subdivisions and multiples: Milli = $\frac{1}{1000}$, Centi = $\frac{1}{100}$, Deci = $\frac{1}{10}$; Deca = 10, Hecto = 100, Kilo = 1000, and Myria = 10,000. In abbreviations the subdivisions begin with a small letter, the multiples with a capital letter. For example:

Millimetre	(.001)	denoted by	mm.
Centimetre	(.01)	"	cm.
Decimetre	(.1)	"	dm.
Metre	(1.)	"	m.
Decametre	(10.)	"	Dm.
Hectometre	(100.)	"	Hm.
Kilometre	(1000.)	"	Km.
1 Centiare	(1 m^2)	"	ca.
Square decimetre		"	dm^2.
Cube metre		"	m^3.
Decilitre		"	dl.
Milligram		"	mg.
Kilogram		"	Kg.

U. S. to METRIC

PRESSURES

1 lb. per sq. in. .... = 0.070307 Kg. per cm^2.
1 lb. per sq. ft. .... = 4.8824 Kg. per m^2.
1 atmosphere (14.7 lbs. per sq. inch)
= 1.03326 Kg. per cm^2.

HEAT UNITS

1 B. T. U. = quantity of heat required to raise temperature of a lb. water, 1° F.
1 B. T. U. = 0.252 cal.
1 B. T. U. per sq. ft. ...... = 2.7124 cal. per m^2.
1 B. T. U. per cu. ft. ...... = 8.8991 cal. per m^3.

UNITS OF WORK

1 foot-pound .... = 0.138255 m. Kg.
33,000 ft.-lbs. per min. (1 engl. H. P.)
= 4562.429 m. Kg.
1 engl. H. P. .... = 1.01387 met. H. P.

QUANTITIES PER HORSEPOWER

1 sq. ft. p. engl. H.P. = 0.091632 m^2. p. met. H.P.
1 cu. ft. " " = 0.02793 m^3. " "
1 U.S. gal. " " = 3.73364 litres " "
1 lb. per " " = 0.447386 Kg. " "

UNIT OF VELOCITY

1 foot per min. .... = 0.016$\frac{2}{3}$ ft. per sec.
= 0.00508 m. per sec.

METRIC to U. S.

PRESSURES

1 gramme per mm^2. = 1.42234 lbs. per sq. inch
1 Kg. per cm^2. ...... = 14.2234 lbs. per sq. inch

HEAT UNITS

1 calorie = quantity of heat required to raise temperature of 1 Kg. water, 1° C.
(1 lb. calorie = quantity of heat required to raise temperature of 1 lb. water, 1° C., or = 1.8 B.T.U. This unit should never be used.)
1 cal. .... = 3.9683 B. T. U.
1 cal. per m^2. .... = 0.36867 B. T. U. per sq. ft.
1 cal. per m^3. .... = 0.112371 B. T. U. per cu. ft.

UNITS OF WORK

1 m. Kg. .... = 7.233 foot-lbs.
1 metric H. P. = 75 m. Kg. per sec. = 542.475 ft.-lbs. per sec., or 32548.49 ft.-lbs. per min. = 0.98632 engl. H. P.

QUANTITIES PER HORSEPOWER

1 m^2. per met. H.P. = 10.9132 sq. ft. per engl. H.P.
1 m^3. " " = 35.8043 cu. ft. " "
1 dm^3. or litre " = 0.035804 cu. ft., or
= 0.267835 U. S. gals. per engl. H.P.
1 Kg. per met. H.P. = 2.2352 lbs. per engl. H.P.

UNIT OF VELOCITY

1 m. per sec. .... = 3.28083 ft. per sec., or
= 196.85 ft. per min.

U. S. to METRIC

CUBIC AND CAPACITY MEASURES

1 cubic inch = 0.016387 l. .... = 16.3871 cm^3.
1 cubic foot ..... = 0.028317 m^3. = 28.317 l.
1 cubic yard .... = 0.76456 m^3.
1 U. S. gallon .... = 3.7854 l.
1 quart (liq.) .... = ($\frac{1}{4}$ gallon) = 0.94636 l.
1 barrel (31.5 gals.) .... = 1.1924 Hl.

WEIGHTS

1 grain (= $\frac{1}{7000}$ lb.) ...... = 0.0648 g. (gramme)
1 ounce (avdup.) .... = 28.35 g.
1 pound (avdup.) .. = 453.592 g. = 0.4536 Kg.
1 ton of 2240 lbs. .. = 1.016 T. = 1016.05 Kg.
1 ton of 2000 lbs. .. = 0.9072 T. = 907.185 Kg.

COMPOUND UNITS

1 grain per cu. foot = 2.28834 grammes per m^3.
1 lb. per cu. foot = 16.01837 Kg. per m^3.

METRIC to U. S.

CUBIC AND CAPACITY MEASURES

1 m^3. = 1 stere = 35.3145 cu. ft. = 1.308 cu. yds.
1 cm^3. .... = 0.06102 cu. inch
1 dm^3. = 1 litre = 61.02338 cu. inches
or .03531 cu. feet
or 1.0567 liq. quarts
or 0.2642 U. S. gallons
1 Hl. = 3.53145 cu. feet
or 0.838636 barrel (31.5 U. S. gal. per bbl.)

WEIGHTS

1 g. (gramme) = 15.432 grains
or 0.035274 avdup. ounces
1 Kg. .... = 2.20462 avdup. lbs.
1 T. (tonne) = 0.9842 ton of 2240 lbs.
or 19.684 cwts., or 1.1023 ton of 2000 lbs.
or 2204.62 lbs.

COMPOUND UNITS

1 gramme per m^3. = 0.436998 grains per cu. ft.
1 Kg. per m^3. = 0.062428 lbs. per cu. ft.

U. S. to METRIC	METRIC to U.S.
LINEAR	**LINEAR**
1 inch......... = 25.4 mm.	1 m. = 39.37 in. or 3.2808 ft. or 1.0936 yds.
1 foot .... = 0.3048 m.	1 mm.......... = 0.03937 inch
1 yard......... = 0.9144 m.	1 cm.......... = 0.3937 inch
1 mile......... = 1.6093 Km.	1 Km. = 1093.61 yds or 0.621 mile
SQUARE	**SQUARE**
1 sq. inch ...... = 6.4516 cm^2.	1 m^2. = 10.7639 sq. ft. or 1.196 sq. yards
1 sq. foot = 929.03 cm^2 or 0.0929 m^2.	1 mm^2...... = 0.00155 sq. inch
1 sq. yard...... = 0.8361 m^2.	1 cm^2...... = 0.155 sq. inch

Fire Insurance.

The following are the requirements of the New York Board of Fire Underwriters for the Installation and use of Kerosene Oil Engines:

Location of Engine.—Engine shall not be located where the normal temperature is above 95° Fahr., or within ten feet of any fire.

If enclosed in room, same must be well ventilated, and if room has a wood floor, the entire floor must be covered with metal and kept free from the drippings of oil.

If engine is not enclosed, and if set on a wood floor, then the floor under and three feet outside of it must be covered with metal.

Oil Feed Tank.—If located inside of building, shall not exceed five gallons capacity, and must be made of galvanized iron or copper, not less than No. 22 B. & S. gauge, and must be double seamed and soldered, and must be set in a drip pan on the floor at the base of the engine.

Tanks of more than five gallons capacity must be made of heavy iron or steel, be riveted, and be located, preferably, underground outside of the building. If there is no space available outside the building for a tank, it may, by written permission from this Board, be located in an approved vault attached to the building, or in a non-combustible and well-ventilated compartment inside the building; but no such tank shall exceed five barrels capacity.

Tanks, irrespective of the method of feed, must not be located above the floor on which the engine is set.

The base of an engine must not be used in lieu of a tank as a receptacle for feed oil. A tank, if satisfactorily insulated from the heat of the engine and approved by the Board, may be placed inside of the base.

In starting an engine, gas only, properly arranged, must be used to heat the combustion chamber.

A high-grade kerosene oil must be used, the flash test of which shall be not lower than 100° Fahr.

Oily waste and rags must be kept in an approved self-closing metal can, with legs to raise it six inches above the floor.

The supply of oil, unless in an approved tank outside the building, or in a non-combustible compartment, as above provided for, shall not exceed one barrel, which may be stored on the premises, provided same is kept in an unexposed location ten feet distant from any fire, artificial light and inflammable material, and oil drawn by daylight only.

A drip pan must be placed under the barrel.

Empty kerosene barrels must not be kept on the premises.

TABLE VI.—TRIALS OF 25 B. H. P. HORNSBY-AKROYD OIL ENGINE, JAN. 4, 1898 (ROBINSON).

Power or Load Factor.	Full Load.	Two-thirds Load.	One-third Load.	No Load.	Maximum Load.
Duration of trial hours.....	3	3	2	1	¼
Revolutions per min. (mean)	202.6	202.4	203	201.5	203
Explosions per minute "	101.3	101.2	100	100.7	101.5
Mean effective pressure (net) lb. per sq. in.......	45.4–43.4	31.2	18.3	6	....
Indicated H. P.............	32.3–31	22.4	13.1	4.28	....
Brake or actual H. P........	26.74	17.96	9.0	0	39
Spent in engine friction, H. P.	5.56–4.26	4.44	4.1	4.28	....
Mechanical efficiency, per cent....................	82.4–86	80	69	......	....
Oil Used in Engine.					
Per hour...............lbs.	19.75	16.75	12	5.75	....
" I. H. P., hour "	0.61–0.63	0.74	0.91	1.34	....
" B. H. P. " "	0.74	0.91	1.3	......	....
Jacket Water.					
Lb. per minute.............	67.5	......	60	......	....
Final temperature (Fahr.)..	138°	130°	132°	142°	138°
Rise in degrees " ..	47°	29°	29°	32°	26°
Equivalent H. P. lost.......	74.8	......	41	......	....
Indicated Pressure lb. per sq. in. above Atmosphere.					
Compression before ignition	60	60	50	55 to 75	....
Explosion pressure.........	168	150	95	......	....
Percentage equivalent of effective heat from oil..					
Useful work at Brake......	18	......	10	......	....
Spent in engine friction...	3	......	4.5	......	....
Shown on indicator diagram	21	......	14.5	......	....
Carried away in jacket water	50	......	45.5	......	....
Balance lost in exhaust gases and unaccounted for	29	......	40	......	....

The day was rainy, with mist and complete saturation of air. The engine was cold when lamp lighted at 10.15 A.M., and started working in five minutes. Observations were made in full load trial at 10.30 A.M.

From "Gas and Petroleum Engines," by Prof. Wm. Robinson, page 710.

TABLE.—RESULTS OF TESTS MADE ON DE LA VERGNE MACHINE CO.'S TYPE FH FUEL OIL AND CRUDE OIL ENGINE.

125 Horse Power, by DR. WALDO. Duration of Trial, 10 hours. Two Oils used: Solar Fuel Oil, about 0.8600 Spec. Grav. and California Crude Oil, about 0.953 Spec. Grav.—*July, 1909.*

	1	2	3	4	5	6	7	8	9	10	Average of Four Hour Runs on: Solar Fuel Oil.	Average of Four Hour Runs on: Cal. Crude Oil.
Amperes	312	312	313	308	212.5	222	269	305.5	305	301.5	311	295
Volts	261	259	258	257	202.5	202	264	264	268	267.5	25[illegible]	268.3
K. W. Switchboard	81.6	81	80.9	79.5	43	44.9	76.6	80.5	82	80.6	80.5	79.9
Electrical H. P	109.3	108.5	108.5	106.5	57.5	60.2	102.5	107.7	109.8	108	—	—
B. H. P., 84% eff. Gen.	130	129	129	126.5	68.5	71.6	122	128.2	130.5	128.5	128.5	127.3
R. P. M.	156	156.5	157	157	158	156	156	156	156	156	156	156
Temperature Cooling Water Inlet, F.	73°	74°	74.5°	74.5°	74.5°	74°	74°	74°	74°	74°	—	—
" " " Outlet, F.	176°	185°	1[illegible]9°	163°	148°	146°	172°	184°	171°	182°	—	—
Consumption " Gallons	292	289	[illegible]88	409	326	xxx	330	386	350.8	xxx	—	—
Cooling Water, per B. H. P., hour	2.25	2.24	2.23	3.23	4.76	xxx	2.70	3.03	2.74	xxx	2.84	2.82
Spec. Grav. Fuel Oil, Beaume.	32°	32°	33.6°	33.5°	33.5°	33°	17°	15°	18°	18°	—	—
Temperature " " Fahr.	86°	88°	92°	94°	94°	94°	90°	122°	116°	116°	—	—
Con. per hr. " " Lbs.	53.2	51.6	48.2	54.2	33.3	32.9	55.8	64.1	67	58.2	—	—
" " B. H. P. hour	0.408	0.422	0.374	0.427	0.487	0.458	0.465	0.505	0.514	0.453	0.408	0.484
Lub. Oil, lbs. per hour	3.47*	3.47*	1.95	1.95	xxx	1.25	0.937	0.75	0.75	xxx	—	—
Cyl. " " " "	1.125	1.125	1.0	1.0	xxx	0.885	0.75	0.937	1.11	xxx	—	—
Lub. " " " B. H. P., hour	0.026	0.027	0.0153	0.015	xxx	0.017	0.007	0.006	0.006	xxx	0.021	0.006
Cyl. " " " " "	0.0096	0.0097	0.0077	0.008	xxx	0.012	0.0065	0.007	0.008	xxx	0.009	0.007
Temp. Resistance, Tank, Cent.	xxx	xxx	xxx	38°	38°	37°	35°	36°	36°	36°	—	—

EXHAUST GAS ANALYSIS.

	1 No Load	2 Full Load	3 Full Load	4 Full Load	5 Half Load
CO_2	1.0	5.7	5.9	6.0	4.0
0	18.8	13.0	12.5	12.3	15.7

* Excess due to oil leakage.

TABLE XVIII.—TESTS OF VARIOUS OIL ENGINES MADE IN EDINBURGH.

ENGINES.	Crossley Bros., Ltd.	Campbell Gas Engine Co.	Campbell Gas Engine Co.	R. Stephenson & Co.	Blackstone & Co.	Blackstone & Co.	Blackstone & Co.	Tangyes, Ltd.	Pollock, Whyte and Waddell.	R. Cundall & Son.
Diameter of cylinder, inches	10	12½	9½	7	6	7	9½	11	10	8¾
Stroke, inches....	18	21	18	12	12	14	18	16	18	15
Price of oil per gal. delivered Edin., pence..	6½	6½	6½	6½	6½	6½	6½	6½	6½	6½
FULL POWER TRIAL:										
Brake horsepower....	15.5	18.93	13.87	3.14	5.21	8.13	12.6	18.06	10.64	8.77
Total oil used per hour, lb.	12.29	22.74	14.74	5.13	4.34	6.78	9.40	14.56	12.31	8.43
Oil per BHP per hour, lb.	.793	1.20	1.06	1.63	.833	.836	.746	.806	1.15	.962
Cost per hour (total), pence	10.08	17.88	11.60	4.20	3.42	5.35	7.42	11.50	10.05	6.86
" per BHP per hour, pence	.65	.94	.83	1.33	.656	.658	.588	.636	.938	.782
HALF POWER TRIAL:										
Brake horsepower....	7.71	10.59	6.73	1.31	2.84	4.84	6.59	9.95	4.69	4.35
Total oil used per hour, lb.	8.00	15.52	7.985	3.78	3.125	4.975	6.75	9.35	10.75	6.496
Oil per BHP per hour, lb.	1.037	1.466	1.186	2.88	1.099	1.03	1.024	.939	2.23	1.57
Cost per hour (total), pence	6.56	12.22	6.28	3.08	2.46	3.92	5.32	7.38	8.77	5.27
" per BHP per hour, pence	.85	1.152	.933	2.35	.865	.812	.807	.741	1.82	1.275
LIGHT POWER TRIAL:										
Total oil used per hour, lb.	4.03	8.23	3.8	4.43	1.69	2.75	3.4	3.375	5.375	4.24
Cost per hour, pence......	3.30	6.47	2.99	3.62	1.33	2.17	2.68	2.67	4.38	3.44
MAXIMUM POWER TRIAL:										
Brake horsepower........	18.01	25.55	14.89	3.14	6.68	10.66	19.7	20.66	19.85	10.54

TABLE XIX.—TESTS OF VARIOUS OIL ENGINES MADE IN EDINBURGH.

ENGINES.	Crossley Bros., Ltd.	Campbell Gas Engine Co.	Campbell Gas Engine Co.	R. Stephenson & Co.	Blackstone & Co.	Blackstone & Co.	Blackstone & Co.	Tangyes, Ltd.	Pollock, Whyte and Waddell.	R. Cundall & Sons.
Duration of trial hours... .. .. —	3.762	4	4	1.266	4	4	4	4	4	2
Time taken to start full load, miuutes..	18¼	31	14	10	11	10	16	15	13½	11
BRAKE HORSEPOWER:										
Circumference of flywheel, effective, ft.	11.322	17.586	15.952	14.214	10.936	14.040	17.529	16.022	15.845	14.372
Load on brake, lb..	224.5	204	147	41	70	98	168	204.75	112	96.25
Spring balance reading, lb.	3	15	11.37	12	8.55	10.3	43.4	18.82	11.5	7.54
Net load on brake, lb.	221.5	189	135.63	29	61.45	87.7	124.6	185.93	100.5	88.71
Revolutions per minute, mean.........	204	188	210	252	256	218	190.3	200.1	220.5	227.7
Brake horsepower....	15.5	18.93	13.87	3.14	5.21	8.13	12.6	18.06	10.64	8.77
INDICATED HORSEPOWER:										
Diameter of cylinder, inches..	10	12.5	9.5	7	6	7	9.5	11	10	8.75
Stroke, inches.	18	21	18	12	12	14	18	16	18	15
Mean effective pressure, lb. per sq. in.	64.52	49.5	—	39	—	—	56	62.2	—	—
Explosions per minute, mean	87.25	76	—	118.5	—	—	81.4	89.75	—	—
Indicated horsepower.....	20.09	24.48	—	5.39	—	—	14.68	21.43	—	—
Mechanical efficiency.................	.771	.773	—	.582	—	—	.858	.842	—	—
OIL CONSUMPTION:										
Description of oil used in trial.........	Royal Daylight	Russolene	Russolene	Royal Daylight	Russolene	Russolene	Russolene	Russolene	Royal Daylight	Royal Daylight
Specific gravity...........	.793	.826	.826	.796	.825	.825	.825	.823	.797	.800
Total oil used, engine and lamps, lb....	46.25	90.97	58.99	6.5	17.375	27.125	37.625	58.25	49.25	16.875
Oil per I.H.P. per hour, lb...........	.611	.928	—	.952	—	—	.640	.679	—	—
" " B.H.P. " " "	.793	1.200	1.200	1.63	.833	.836	.746	.806	1.15	.962

3918

INDEX

Abel oil-tester........... 90
Actual horse-power....... 63
Air compressing, horse-power required.......125
Air-compressor at different altitudes..........129
Air-compressors......123, 204
Air inlet choked......... 77
Air-inlet valve..12, 39, 57, 61, 78, 145, 172, 175
Air-inlet valve, automatic.........12, 77, 156
Air-pump 13
Air suction, noise of.... 122
Air-suction pipe.......... 78
Air-supply (Campbell)...151
Air-supply (Crossley)....149
Air-supply (Priestman)..152
Analyses, oil............ 232
Asbestos 58
Assembling oil engines... 53
Atmospheric line......70, 71

Balance weights......... 30
Balancing crank-shaft.... 28
Balancing fly-wheel....... 30
Balancing formula........ 29
Barker Engine...........197
Bates, F. H.............221
Bearing caps............ 55
Bearings, crank-shaft..40, 158
Bearings, outside.........172
Bearings, pressure on.... 40
Bearings, scraping in..... 54
Beau de Rochas Cycle, 15, 16, 76, 215
Beaumont crude oil......232
Belt centres..............115
Belt, link...........113, 115
Belt, loose...............115
Belt, size of.............116
B. H. P., to calculate.... 65
Brake, attaching......... 64
Brake, horse-power..23, 63, 64
Britannia Co.'s Engine...192

Campbell, governing, 13, 151, 175
Campbell oil engine described 172, 250
Campbell starting........150
Cams 37
Cams, setting............ 60
Circulating water-pipes... 97
Clerk, Dugald........... 87
Clutches, friction.........137
Clutches, friction, advantages of..............137
Clutches, friction, B and C type..............138

Coal oil.......... 1
Combustion, bad......89, 153
Combustion, complete.....90
Compression (Diesel)...5, 25
Compression in crank-chamber 180
Compression, increasing.. 79
Compression, irregular.... 19
Compression line......76, 78
Compression pressure, 22, 25, 164
Connecting-rod bearings 56
Connecting-rods.......31, 32
Connecting-rods, diameter 33
Connecting-rods, phosphor bronze 31
Cooling towers...........100
Cooling water........19, 201
Cooling water-tanks...... 96
Copper ring.............. 58
Cost of installation.....209
Crank-pin............42, 175
Crank-pin, dimensions.... 42
Crank-pin, size of........ 40
Crank-shaft 26
Crank-shaft, balancing... 28
Crank-shaft bearings..42, 158
Crank-shaft, strength of.. 26
Crossley engine described.168
Crossley engine, portable.203
Crossley governing.......171
Crossley measuring device.168
Crossley starting.........148
Crude oil vaporizer..220, 231
Crude oil, Beaumont.....232
Crude oil, California.....239
Cundall engine described.172
Cycles, different, discussed 18
Cyclic variation.......35, 36
Cylinder clearance........ 25
Cylinder cover........... 25
Cylinder lubricating oil..140
Cylinder lubricators...... 58
Cylinder, two parts...... 57
Cylinders, different types 22, 24

Defective air-supply.....164
Defective oil-supply..... 164
Denton, Prof............218
De la Vergne engine 129, 185, 226
Diagram, analyzing....... 77
Diagram, good working.. 76
Diesel governing.........217
Diesel heat balance......218
Diesel motor..........5, 210
Diesel starting...........210
Direct-connected engine and dynamo......... 117
Direction of rotation, reversing 154
Distance pieces 55
Draining, water...........104
Dynamo fly-wheel.........115
Dynamometer or brake... 64

"Economist" Retort......221
Effective horse-power.... 63
Efficiencies, thermal, compared 87
Efficiency, increase of.... 83
Efficiency, mechanical..23, 86
Efficiency, thermal........ 86

Electric igniter....5, 15, 152
Electric lighting plant, installation of..........113
Electric lighting, portable.200
Engine (Campbell).....172
Engine (Cundall)........172
Engine frame............ 43
Engine (Hornsby-Akroyd) 140, 182, 211
Engine, ideal heat....... 21
Engine (Mietz & Weiss).178
Engine, portable..........200
Engine (Priestman).....175
Engines (Barker)........197
Engines (Britannia Co.'s) 192
Engines (Crossley)......168
Engines (Crossley portable) 202
Engines (American Oil Engine Co.'s)........194
Engines driving dynamos..111
Engines, electric lighting.. 46
Engines (Fairbanks-Morse) 225
Engines (Traction)......205
Engines, knocking...159, 164
Engines, large size......206
Engines (Mietz & Weiss portable) 203
Engines, regulation of....117
Engines, running, general remarks 153
Engines, running, light...145
Erecting oil engines...... 53
Exhaust bends........... 41
Exhaust, choked......... 83
Exhaust gases...90, 153, 165
Exhaust line..........76, 83
Exhaust silencers........100
Exhaust temperature.....110
Exhaust valve........... 13
Exhaust valve, opening of. 76
Exhaust washer..........101
Expansion line........76, 81
Explosion 20
Explosion in silencer....166
Explosive mixture....10, 15

Fairbanks Morse Engine.225
Filter oil........49, 146, 160
Fire insurance............244
Flashing point of oil..... 1
Flashing point to test.... 90
Flickering of incandescent lights 119
Fluctuation in speed..... 37
Fly-wheels............35, 119
Fly-wheels, energy of.... 35
Fly-wheels for dynamo..115
Fly-wheels, formula for.. 37
Fly-wheels, keying on.... 57
Fly-wheels, peripheral speed 37
Foundations 113
Four-cycle 15
Frame, engine............ 43
Friction-clutches 137
Friction-clutches, advantages of..............137
Friction-clutches, B and C type...............138
Frost, provision for...... 99
Fuel-consumption test.... 87
Fuel injection....10, 165, 216

Fuel, injection of........ 53
Fuel oil-tank.....13, 49, 168, 172, 174, 176, 177, 180
Fuels...............230, 236

Gases, exhaust........... 90
Gear, skew............... 43
Gear, spur...........43, 160
Gear, starting...........106
Governing (Campbell), 13, 151, 175
Governing (centrifugal), 15, 168, 171, 172, 175
Governing (Crossley)...171
Governing devices.....44, 48
Governing (Diesel)......217
Governing (Mietz & Weiss) 179
Governing (Priestman), 15, 176
Governor, hit-and-miss type 45
Governor hunting........148
Governor parts, renewing.160
Governor, pendulum type. 45
Governor, Porter type....180
Governor, Rites......45, 189
Gravitation (fuel)....12, 175
Gravitation system....... 96

Heat losses.............. 22
Heat, utilization of waste..107
Heated air.............. 11
Heat balance............ 87
Heat balance (Diesel)...218
Heating lamp.......8, 11, 12
Heating lamp instructions.141
Horizontal and vertical types 50
Hornsby-Akroyd, instructions for running, 140, 182, 211
Hornsby-Akroyd, method of vaporizing........ 9
Hornsby-Akroyd oil supply 10, 180
Hornsby-Akroyd Traction Engine 205
Hornsby-Akroyd vertical type 187
Horse-power 63, 66

Ice and refrigerating machines 133
Igniter, electric....5, 15, 152
Igniter (Hornsby-Akroyd) 2
Igniters 2
Igniters (flame).......... 2
Igniters, heating.......... 61
Ignition 140
Ignition (electric).......2, 7
Ignition (high compression)..............2, 215
Ignition (hot surface) 2, 7, 10
Ignition (hot tube), 2, 7, 11, 148, 151
Ignition line............. 76
Ignition line, late........ 80
Ignition line, too early.. 79
Ignition, regulating....... 80
Ignition, retarding....... 81
Impulse on piston........ 17
Incandescent lights.......116

Incandescent lights, flickering of..............119
Indicated horse-power..23, 66
Indicator attaching to engine 71
Indicator cock........... 66
Indicator, Crosby........ 67
Indicator diagram, 75, 170, 174, 184, 218
Indicator diagram, light spring 88
Indicator, diagram measuring 73
Indicator in place........ 64
Indicator, left or right hand 70
Indicator reducing motion 71
Indicator springs........ 69
Ingredients for foundations 113
Installation, Cost of......209
Instructions for running Hornsby-Akroyd 140
Instructions for running oil engines...........139
Insurance, Fire...........244

Johnston oil Engine.....191
Junk rings............... 55

Knocking in engine..159, 164

Large size Engines......206
Leakage in crank-chamber 19
Leakage of piston-rings, 61, 78, 165
Leakage of valves........ 78
Leakage of water into cylinder 63, 166
Lights, incandescent......116
Line, atmospheric.....70, 71
Line, compression.....76, 78
Line, exhaust..........76, 83
Line, expansion.......76, 81
Link belt............113, 115
Loose belt..............115
Loss of power...........165
Lubricating cylinder oil..140
Lubricators, cylinder..... 58
Lucke & Verplank Vaporizer 8

Magneto 4
Measuring device (Crossley) 168
Mechanical efficiency, 23, 51, 86
M. E. P..............67, 81
M. E. P. regulated....... 47
Method of vaporizing (Crossley) 11
Method of vaporizing (Campbell) 12
Method of vaporizing (Hornsby-Akroyd) .. 9
Method of vaporizing (Priestman) 13
Method of governing (Campbell) 175
Method of governing (Diesel) 217
Method of governing (Mietz & Weiss).....178

Method of governing (Priestman) 176
Metric measures.........241
Mietz & Weiss engine described,
52, 128, 178, 211, 203
Mietz & Weiss engine governing 179
Mixture oil, vapor and air 14
Moore, C. C. & Co...206, 222
Motor, Diesel.........6, 210
Multi-cyclinder engines... 51

Norris, William......... 26

Oil, Beaumont..........232
Oil, California..........239
Oil, crude...............231
Oil cylinder, lubricating...140
Oil engines, driving dynamos 111
Oil engines, instructions for running..........139
Oil filter.........49, 146, 160
Oil injection..........10, 216
Oil inlet................ 12
Oil measurer (Crossley).. 11
Oil-pump.........9, 143, 172
Oil-pump, testing.........147
Oil supply (Campbell)...151
Oil supply (Crossley)....171
Oil supply (Diesel)......215
Oil supply (Hornsby-Akroyd) 182
Oil supply, limiting...89, 164
Oil supply (Mietz & Weiss) 177
Oil-supply pipes..57, 61, 146
Oil supply (Priestman).. 15
Oil-supply pump.........178
Oil sprayers.............. 13
Oil, viscosity of......... 93
Otto cycle............15, 76
Otto patent.............. 19

Paraffin (Scotch)...... 1
Petroleum 1
Petroleum (crude)
2, 220, 231
Petroleum. See Tables.
Pipe, air-suction......... 78
Piston 33, 35, 41, 153
Piston, blowing..........165
Piston, fitting............ 55
Piston lubrication.50, 158, 170
Piston-rings,
34, 55, 56, 154, 158, 159
Piston speed...........22, 34
Piston, taking out........158
Piston, water-jacketed.... 34
Planimeters 72
Planimeters, directions for using 74
Plants, pumping..........131
Portable engines.........200
Portable engines, construction of..........200
Port openings............ 39
Pressure of explosion.... 20
Pressure on bearings.... 40
Priestman engine.....14, 175
Priestman, governing.15, 176
Priestman, starting......152
Priming cup (Crossley)..148

Processes in cylinder of engine 59
Products of combustion.. 18
Pump, oil-supply......... 49
Pump, water-circulating... 98
Pumping-plants......130, 131
Pumps, efficiency of......133
Pumps, horse-power required 132

RADIATORS for cooling.... 99
Ratio, air and oil vapour.. 7
Refrigerating machines...133
Refrigerating machines, horse-power required..136
Refrigerating machines, rating of.............133
Regulation of engines....117
Retort, "Economist".....221
Reversing direction of rotation 154
Rhumkorff coil........... 5
Rings, junk.............. 55
Rings, piston, 34, 55, 56, 154, 158, 159
Rites governor.......45, 189
Robinson, Wm...178, 220, 250
Running oil engines......139

SELF-STARTER 105
Self-starter (Hornsby-Akroyd) 105
Silencers, exhaust........100
Simplicity of construction 22
Single cycle.............. 16
Skew-gear 43
Specific gravity...1, 232, 235
Speed counter (Hill).... 85
Speed, regulation of......154
Sprayer 13, 14
Spray holes..............147
Spur gear............43, 160
Starting 7, 11, 215
Starting (Campbell type) 150
Starting (Crossley type)..148
Starting (Diesel motor)..215
Starting, difficulties of 61, 143, 164
Starting gear.............106
Starting (Hornsby-Akroyd) 142
Starting (Priestman type).152
Starting valve...........215
Straight line principle....175
Stroke, ratio............. 26
Suction line.............. 76

TACHOMETERS 83
Tachometers, portable.... 84
Tank 49
Tank, fuel consumption.. 64
Tank, water..............141
Temperature of cooling water 81, 100
Temperature, exhaust....110
Test (Diesel)............220
Test (Hornsby-Akroyd) 247, 250
Test (Priestman)........178
Test (Various).......247-251
Testing compression..61, 164
Testing flash-point...90, 232
Testing fuel consumption 87
Testing new engine...... 59

Testing, object of........ 59
Testing oil-pump.........147
Testing sprayer.......... 61
Testing water-jackets..... 63
Thermal efficiency....86, 218
Timing of ignition.......162
Traction Engine.........205
Tube igniter........3, 5, 163
Two-cycle system..15, 44, 177

VALVE, air and exhaust,
39, 57, 145, 158, 216
Valve, back pressure.....146
Valve by-pass........45, 180
Valve closing-springs..... 39
Valve exhaust opening... 60
Valve, lift of............ 78
Valve mechanisms........ 43
Valve, overflow, oil......146
Valve starting...........215
Valves 41
Valves and valve-boxes.. 38
Vapor inlet-valve..11, 12, 150
Vaporizer 7
Vaporizer, advantages of.. 8
Vaporizer (Campbell).... 5
Vaporizer (Crossley)..11 150
Vaporizer, difficulties of. 9
Vaporizer heated by exhaust 14
Vaporizer, heating....61, 152
Vaporizer (Hornsby-Akroyd) 9
Vaporizer (Priestman)... 13
Vaporizer, to heat.......141
Vaporizer valve-box......145
Vaporizer, water-jacketed.141
Vaporizers, crude-oil.....220
Vertical engines.......... 51
Vibrator 6
Viscosity of oil.......... 93

WASHER, exhaust........101
Waste heat, utilization of..107
Water-circulating pipes... 97
Water-circulating pump.. 98
Water cooling...........201
Water draining..........104
Water in exhaust pipe....104
Water-jackets.........57, 212
Water leakage...........166
Water injection.......... 52
Water space............. 25
Water-tanks, capacity of.. 96
Water-tanks, cooling..96, 141
Worm-gear 43, 160